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ABSTRACTS NO.

26,703 — 26,777

THE BRITISH SHIP RESEARCH ASSOCIATION

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53

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TABLE OF CONTENTS

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ABSTRACTS *from Current Technical Literature*

The following Abstracts purport to be fair summaries of the articles, but the Association does not accept responsibility for statements made in the originals, nor does it necessarily agree with their contents.

The standard form of reference to the source of each Abstract is: Title of Periodical or Publication (abbreviated according to the list on pp. 3-19 of B.S.R.A. Journal for January 1968), volume number (in heavy type), year, and page number, followed by the date of issue where appropriate. The length of the article and other bibliographical details are also indicated.

SHIP RESISTANCE AND FLUID MOTION

26,703 **A Report on Design Improvements to the Blohm & Voss " Basic Pioneer " Liberty Replacement.** GALLIN, C. *Shipbuild. Shipp. Rec.*, **111** (1968), p. 327 (8 Mar.) [4 pp., 3 tab., 6 graphs, 5 diag., 4 phot.]

This is an English version of the German article summarised in Abstract No. 26,303 (Apr. 1968).

26,704 **Turbulent Flow of Dilute Aqueous Polymer Solutions.** GOREN, Y., and NORBURY, J. F. *A.S.M.E., Paper No. 67-WA/FE-3, presented* 12-17 Nov. 1967 [9 pp., 17 ref., 2 tab., 11 graphs, 5 diag., 1 phot.]

This paper summarises some of the research into the effect of polymer additives on turbulent shear flow, which was conducted at the University of Liverpool between Oct. 1964 and Oct. 1966. It contains a description of the research, together with the principal results, discussion, and conclusions.

The work was devoted to a detailed examination of the mechanism of a particular flow by gathering information on friction drag, velocity distribution, concentration distribution, and correlation with Reynolds number and polymer concentration level. The particular flow chosen was the fully developed turbulent flow of Polyox WSR-301 solutions in a 2-in dia. pipe.

A maximum drag reduction of 71 % was obtained at a Reynolds number of $1{\cdot}5 \times 10^5$ for solutions having polymer concentration of 10 weight p.p.m. The drag-reduction effect occurred only above some " critical " Reynolds number which was independent of concentration.

The polymer additives were found to influence the flow in the neighbourhood of a solid boundary. In this zone of the flow, the eddy viscosity was found to be much lower than that of water. In the absence of a boundary, as in free jet flow, the polymer additives had no effect on the flow characteristics. The experiments showed for the first time that the polymer molecules were uniformly distributed across the pipe diameter under all turbulent flow conditions investigated. A method of determining polymer concentration was devised for this purpose.

PROPELLERS AND PROPULSION

26,705 **A Rig for Measuring Model Contra-Rotating Propeller Forces, and Test Results.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 654 (10 May) [1½ pp., 1 diag., 1 graph, 1 phot.]

The article describes apparatus used in the self-propelled model tests carried out in the Vickers Ship Model Tank, St Albans, on a 200,000-ton d.w. tanker, as mentioned in Abstract No. 26,379 (May 1968). Developed at the Vickers tank to an order placed by Stone Manganese Marine, it is the first apparatus available in the U.K. which makes it possible to measure independently the forces on the separate components of a pair of model contra-rotating propellers. Torque and thrust on each member of the pair are measured by two separate dynamometers.

A full, illustrated description of the apparatus is given. Mention is made of the plain journal bearings by which the two tail shafts, one (solid) inside the other which is tubular, are supported by the stern tube. These bearings are made from a PTFE type of plastic which has extremely low friction, and are lubricated by slow leakage of water from the tank into the model.

Test results obtained with this equipment are given for a 710-ft container ship with four different propeller arrangements. QPC is plotted to a base of speed for (1) a normal screw propeller, (2) c.r. propeller based on Glover's theory, (3) c.r. propeller based on Lerbs' theory, and (4) a single propeller in a c.r. propeller aperture. As compared with (1), (2) shows an improvement of about 10·5% at 19 knots and 8·5% at 24 knots, and (4) is inferior by about 1·5% over the whole speed range; (3) is somewhat inferior to (2), but still markedly better than the single-screw conditions (1) and (4).

SHIP PERFORMANCE, STABILITY, AND MANOEUVRABILITY

(*See also Abstracts No.* 26,712, 26,713 *and* 26,723)

26,706 **Ships' Stabilisers: Fins and Tanks.** Rorke, J. *Shipbuild. Shipp. Rec.*, **112** (1968), p. 185 (9 Aug.) [3½ pp., 5 diag., 4 graphs]

The Author deals only with roll stabilisation, since it is not practicable to damp effectively the other two motions, pitch and heave, which are, with roll, the most important as regards the safety of the ship and the comfort of passengers and crew. He summarises the standard theory of rolling motion and rolling stability of a ship, and the effects of stabilisers; and shows that, for the stabilising torque to be equal to the sea rolling moment, the torque can be expressed by a formula which includes the maximum slope of an equivalent sinusoidal wave. This slope, or angle, defines the " power " of the stabiliser.

The power of fin stabilisers varies approximately as the square of the speed, being zero at zero speed. These stabilisers are usually designed to have a power of 5° at the ship's service speed. With this power, a fin stabiliser will reduce a ship's roll at resonance from 30° out to out to as little as 3°. This residual roll is well above the minimum that can be sensed by the gyroscopic unit that interprets the roll and passes the necessary signals to the fin operating gear.

Fin stabilisers are more effective and efficient than any other system for stabilising ships that operate at high sustained speeds, but their stabilising power at low speed is negligible. On the other hand the tank stabiliser, which is independent of ship speed, will give 2° to 3° of stabilising power from zero to full speed. Basically, the tank stabiliser consists of a tank or tanks mounted athwartships, preferably at the position of maximum beam. The tanks can be either of the open-flume type or, as in the Muirhead-Brown type (see Abstract No. 25,317, May 1967), connected by a water-transfer duct at the bottom and an air-transfer duct at the top. Valves in this air duct, operated by a roll-sensing element, control the flow of water between the tanks. The Author describes the way in which this system operates and draws attention to its three main advantages, namely, that the optimum roll reduction is achieved at or near resonance of the ship; that the residual roll (about 7°) remains practically constant for all sea periods; and that changes in the GM of the ship can be allowed for by the control. The system provides a greater roll reduction at all wave periods and for all sea conditions than any other tank-stabilising system. There is, however, a technical limit to the stabilising power of any tank stabiliser due to the loss of GM caused by free-surface effects. In many ships the limit is about 2° to 3°.

Although less effective than a fin stabiliser at top or service ship speed, the tank stabiliser has the advantage of being cheaper. Investigations are in progress on the use of both systems in combination, and the Author believes that it will be possible to produce such a combination at a cost about equal to that of the fin system alone.

The Author describes briefly a method of presenting sea-trial results which will help shipowners to decide whether or not to fit stabilisers in ships on the routes they cover. The method is based on statistical probability and provides a more useful appraisal of the merits of a stabilising system than a quoted roll reduction at resonance. Briefly, a ship is allowed to roll, both stabilised and unstabilised, for about 100 cycles in each of a number of different weather conditions; and the results are plotted to give curves on a base of angle of roll, out to out, the ordinate of each point being the percentage number of rolls that exceed the roll angle represented by its abscissa.

STRUCTURAL DESIGN AND ITS APPLICATIONS

26,707 **Developments with Respect to Ship's Strength.** ROREN, E. M. Q. *Shipbuild. Shipp. Rec.*, **111** (1968), p. 785 (7 June) [5½ pp., 10 diag., 5 graphs]

In the past, developments in shipbuilding have been largely empirical, based on experience at sea. This slow process is unsatisfactory for problems that arise from the very rapid increase in merchant-ship size that has taken place in the past decade or so; and recently investigations of a more fundamental nature have been started in connection with the design and construction of tankers in the 500,000 to 1,000,000 tons d.w. class. The main aspects of these studies of the loading and strength of ship structures are the following:

(1) Waves and wave-induced loads.
(2) Structural response.
(3) Structural optimisation, i.e. minimisation of the steel weight.
(4) Hull materials, with special reference to their properties as regards welding, brittle fracture, and fatigue. The use of yield-controlled high-tensile steel is increasing, but has led to new fatigue problems.
(5) Rational understanding of structural safety, i.e. statistically-based assessments of the margins of safety against specified types of damage.

The Author deals mainly with items (1) and (2).

Waves and wave-induced loads—The main purpose here is to formulate statistical long-term distributions for variables significant for the scantling process, i.e. distributions that will apply for the lifetime of the ship. The elements in the calculation include the description of the sea based on the Pierson-Moskowitz wave spectrum, which is described and illustrated by a graph, and the ship's response, both short-term and long-term. Short-term response is connected largely with definite circumstances such as specified weather conditions; long-term response is concerned with the expected largest values in the lifetime of the ship.

Structural response—The calculation of stresses and deformations is being greatly facilitated by the use of computers and suitable programs. The two main basic methods of calculation are the application of two- and three-dimensional frame programs and of finite elements. These are explained briefly, their use being exemplified by the design of a web frame for a tanker.

The Author then discusses some special strength problems. The longitudinal strength is mainly assessed on the basis of reasonable safety with regard to collapse (i.e. the yield moment of the hull cross-section), brittle fracture, and fatigue. Collapse is based on plastic considerations, and brittle fracture on static considerations with regard to loads and thermal conditions. Fatigue presents problems about which considerable uncertainty still prevails. One example is connected with the use of " yield-controlled " steels, whose fatigue strength is now known not to be as high in relation to the static strength as was previously thought.

Local strength members are dimensioned in accordance with the effects that any possible damage would have on the structural integrity. The designer has to assess the effects of both service loads and extreme loads. The Author amplifies these remarks by giving examples that arise in various types of ship. For tankers he discusses first the design of the wash bulkheads, which not only reduce the danger of standing waves in the tanks and moderate any internal wave impact, but also play an important part in supporting bottom and deck structures. They are at present analysed by the finite-element method, which enables the distribution of stresses and deformations to be calculated with high precision.

Other problems discussed in connection with local strength in tankers are: the design of web frames; the distribution of stress in corner and bracket areas; structural stability around openings for longitudinals and other openings in highly-stressed areas; the web-plate thickness of deep transverse girders; the docking of large ships; and the inspection of large tanks.

Problems associated with bulk carriers and container ships are also mentioned. The deck structure of a bulk carrier is often designed in such a way that a centreline girder is only partly effective in the longitudinal strength, but the extent to which the girder shares in the hull section modulus can be determined fairly simply. Important factors in the calculation are the stiffness of the transverse deck strips and the terminal conditions of the deck girders. Also discussed are the manner in which a large part of the load on the double bottom is transferred to the ship's sides by the transverse bulkheads; and the design of side-tank frames.

Container ships necessarily have extremely broad hatches, often

85–90% of the ship's breadth. The torsional stiffness of the hull could therefore be so small that large warping motions would affect the tightness of the hatches. Methods of counteracting these effects are briefly mentioned. Present design methods are based on the general torsion theory of thin-walled structures, but the application of finite elements is being studied.

Finally, the Author discusses the avoidance of vibrations in superstructures and afterbodies. It is essential that the stiffness of the superstructure and its attachment to the hull are such that its lowest natural frequency is above any excitation frequency, e.g. frequencies due to the propeller blades or unbalanced engine forces and moments. The principal type of superstructure motion to be avoided is illustrated; it is, in effect, an alternating rise and fall of each end of the superstructure. As regards tank structures in the afterbody, it is important to avoid vibrations that can lead to crack propagation, and the natural frequencies of the structures must be suitably controlled.

26,708 **The Torsional Behaviour of Ships with Large Hatch Openings: Some Further Experiments. Part 2.** Wilde, G. de. *Shipp. World & Shipb.* **161** (1968), p. 423 (Feb.) [3½ pp., 1 tab., 12 diag.]

Part 1 of this article is covered by Abstract No. 26,530 (July 1968). Part 2 is based largely on earlier work by the same Author (see Abstract No. 25,471, July 1967). It presents, in the form of appendices to Part 1, theoretical methods for: (1) Evaluating the influence of deck structure inside the line of openings. (2) Idealising the actual cross-section to an open single-wall cross-section and calculating the torsional rigidity under unrestrained warping. (3) Calculating the warping rigidity and other sectional properties.

26,709 **The Reduction in the Strength of Hull Structures Caused by Making Openings and Subsequently Closing Them [in the Course of Repair Work]** (in Russian). Barabanov, N. V., and Chibiryak, I. M. *Sudostroenie*, No. 9 (1967), p. 10 (Sept.) [4 pp., 4 ref., 1 tab., 2 graphs, 2 diag.]

Especially in ship repair work, it is often necessary to make openings in a hull which are subsequently closed or suitably reinforced. Making such openings causes redistribution of stresses, and residual stresses remain after the openings have been closed. These residual stresses may, when added to the overall hull-bending stresses, lead to cracking in parts of the structure. Replacement of large parts of the shell plating and side framing is most often occasioned by general material wastage. In northern waters ice damage is another important cause; some particulars are given of extensive repairs to the timber carrier *Sibirles* (see Abstract No. 24,872, Dec. 1966) over about a third of her length on both sides. On the starboard side, the cut-away region included the sheer strake and part of the bilge strake, and much of the framing was cut away.

The Authors consider the problem of calculating the stress condition of a hull when a large amount of plating and framing is being renewed; such calculation is necessary for evaluating residual stresses, strength, and sagging deflection, this last being particularly important for relatively flexible hulls. A hull from which strakes of side plating have been removed on one or both sides is treated as a composite beam made up of two beams of varying cross-section, representing the deck and bottom portions respectively. These portions are joined by the frames, which are considered to be fixed in their transverse planes, but elastically compliant in shear. The ends of the deck and bottom portions are considered to be rigidly joined but free to move as a whole. Equations are given for bending moment, direct stress in the deck and bottom portions, and deflection, over the length of the composite beam. In cases where the ship has a long parallel body, and the same length and (constant) depth of plating is removed on both sides, these equations are easily solved.

The Authors have calculated the changes in strength of a ship 97·4 m (319·6 ft) in length (L), 14·4 m (47·2 ft) in breadth, 6·9 m (22·6 ft) in depth, and having a simple midship section (shown in a sketch), when a strake of plating is removed from both sides over lengths of ½ L, ¼ L, and ⅛ L (centred on the midship section), and also when the plating is removed from one side over ½ L. All the frames are retained. The results are presented graphically, the stresses in the deck and bottom portions of the composite beam being shown as a function of the length of plating removed. Deflections can be 3–4·5 times greater than those of the intact hull; however, when plating is removed on one side only the hull behaves as though it were an integral beam.

All methods of reinforcing openings assume that, during the installation of the reinforcement, the structure is in an unstressed condition. It is therefore necessary to ascertain what additional stresses are caused by installing reinforcements in a stressed structure. It is shown how this can be done using mathematical elasticity theory; formulae are derived for the stress distributions, taking account of any difference between the stiffness of the original hull plating and that of the closure or reinforcement. As an example, the case is examined of a circular hole being cut in a stressed plate and then filled in with a plate of greater stiffness. It is shown that the stresses set up depend not only on the loads, but also on the overall bending stresses, operative at the times when the opening is made and closed. A table illustrates the variation of the stresses set up around such a patch with stiffness ratio, and with the relation of the overall stress (in the plating concerned) during repair to that during service. It appears from this table that the stresses around the patch can be considerably reduced by suitable selection of the thicknesses of the repair plate and of the basic hull plating. In order to limit such stresses, it is necessary to ensure that the stresses caused by overall hull bending in the region of the hull under repair are as small as possible.

WELDING AND OTHER METHODS OF CONSTRUCTION

26,710 **Plasma-Arc Welding Processes.** PRIVOZNIK, L. J. *A.S.M.E., Paper No 67 DE 46, presented* 15–18 May 1967 [7 pp., 3 ref., 7 tab., 1 graph, 5 diag., 12 phot.]

The plasma-arc welding process can be regarded as a modification of the conventional tungsten inert-gas (TIG) process, the main differences being in arc configuration and in the function of the inert gas. The principles and applications of high-current (150–300 A) and low-current (0·1–15 A) plasma welding with "transferred" or "non-transferred" arc are described. High-quality single-pass welds can be produced in material thicknesses ranging from 0·5 to 0·001 in. The welds can be made at higher speeds and lower overall costs than welds made in comparable thicknesses of material with the TIG process; this is illustrated by data. Typical welding conditions are given for various steels, nickel and copper alloys, and titanium, in the thickness range mentioned. Edge preparation for single-pass welds is a square butt; information is given on edge preparation for two passes.

26,711 **Appliance for Clamping during Welding** (in German). *Werkstatt und Betrieb*, **100** (1967), p. 836 (Nov.) [½ p., 1 diag.]

A brief description is given of the Elbe clamping press, which is designed to position and hold ferrous parts together for welding. It is intended for use, in shipbuilding and other industries, when welding such parts as frames, girders, and stiffeners to flat material lying in the horizontal position; the frame, etc., can be from 150 to 750 mm, i.e. 5·9 to 29·5 in, deep. The correct relative position of the parts in the horizontal plane is obtained through handwheel control; the parts are held together by an arrangement of two electromagnets (each capable of a force of 3 tons), and a hand-hydraulic pressure-pad (capable of 4 tons force). Welding of long parts entails either moving the press from time to time, or using two presses simultaneously.

The article gives further details of this clamping press, and includes a general-arrangement drawing. The press is produced by Hans Koch (Werkzeugmaschinen), of Geesthacht (near Hamburg).

SHIPBUILDING (GENERAL)

26,712 ***Fotini L* Bulk Carrier for Livanos Interests.** *Shipp. World & Shipb.*, **161** (1968), p. 403 (Feb.) [7 pp., 2 tab., 7 graphs, 9 diag., 4 phot.]

The 74,200-ton d.w. *Fotini L* is the largest ship yet built by the Hakodate Dock Co. Ltd, Hokkaido, Japan. She is owned by Elcapitaine Inc., of Monrovia, an affiliate of Ceres Hellenic Enterprises Ltd, Piraeus, and is the first of six ships (two of 74,200 and four of 25,000 tons d.w.) on order at this yard for Livanos interests. She can transit the Panama Canal. Automation of the main and auxiliary machinery allows the engine room to be unattended at night. The ship conforms to the American Bureau of Shipping classification A1 E & AMS as a bulk carrier strengthened for heavy cargoes with Nos 2, 4, 6, and 8 holds empty. Any type of grain with a stowage factor of 45–70 cu ft/ton can be carried to the full capacity of the nine holds.

The principal particulars are:

Length, o.a.	858·1 ft
b.p.	800 ft
Breadth, moulded	106 ft
Depth, moulded	60 ft
Design draught, moulded	38 ft
Deadweight	74,203 tons
Design load displacement	76,271 tons
Full load displacement on summer load line	90,153 tons
Gross tonnage	36,365

Cargo capacities, grain	. . . .	2,961,329 cu ft
bale	. . . . .	2,931,623 cu ft
ore (in Nos 1, 3, 5, 7, and 9)	. . . .	1,606,580 cu ft
Ballast capacity	. . . .	1,236,584 cu ft
Service speed	. . . .	16·25 knots
Cruising range	. . . .	25,000 miles

The layout is of all-aft type, with poop and forecastle. The bow is of combined cylindrical and bulbous form; it extends 11·4 ft forward of the FP, and its sectional area at the FP is 9·6% of the ship's midship section. The round-plate stem is sharply raked above water. A streamlined balanced rudder is fitted below the cruiser stern. All hatches have MacGregor fore-and-aft sliding steel covers operated by electric winches. Nos 1, 6, and 9 holds and hatches are slightly smaller than the remainder. Upper and lower wing ballast tanks are provided. In the cargo section, the intermediate transverse bulkheads are corrugated, those bounding No. 6 hold horizontally and the others vertically.

When designing the ship, some eighteen possible loading or ballast conditions were examined; these involved cargo stowage factors ranging from 20 to 48 cu ft/ton. Four of these conditions (heavy cargo, bauxite, coal, ballast) are shown in diagrams, together with the resulting bending-moment and shear-force curves and relevant hydrodynamic data. Departure and arrival conditions when carrying grain with stowage factors ranging from 45 to 70 cu ft/ton were also studied, and the results are summarised in a table. These studies are based on loaded operation at a freeboard draught of over 44 ft, authorised because of compliance with the stringent reduced-freeboard conditions (which are outlined) of the 1966 Load Line Convention.

There are eight sets of 10·5-ton automatic-tension mooring winches, two sets of 8·4-ton winches for the hatch covers, and a 65-ton anchor windlass. There is no cargo-handling gear, but a mobile bucket-crane (stowed in the forecastle) is carried; this is used mainly for cleaning out the holds.

The accommodation, which is air-conditioned and finished in plastic-overlaid plywood, provides single-berth cabins for a total complement of 41, but the ship will normally operate with a crew of 28. There is a swimming pool on the bridge deck, and a lift between this deck and the engine room.

The main engine is a Uraga-Sulzer 9RD90, with a normal service output of 18,630 b.h.p. at 115 r.p.m. and a rated output of 20,700 b.h.p. at 119 r.p.m. The engine can be controlled from the bridge or from a port-side machinery control room in the engine room. It drives a five-bladed 21-ft diameter propeller. There are two 600-kW Diesel-alternators and one of 300 kW; they have medium-speed Daihatsu engines. When at sea, power is obtained from a 620-kW turbo-alternator powered by a Uraga steam turbine developing 920 h.p. at 1,800 r.p.m. Steam is supplied by a Cyclotherm auxiliary boiler with an output of 5,940 lb/hr at 114 lb/sq in, and an exhaust-gas boiler rated at 13,750 lb/hr at 114 lb/sq in. Most of the auxiliary equipment is of Japanese manufacture, but Graviner (Colnbrook) Ltd have supplied the oil-mist detectors and the scavenge-duct fire detector for the main engine.

The conditions and results of speed trials and manœuvring trials are given, partly in tabular and graphical form. The trials included crash-stop astern and ahead tests, stopping inertia tests, and turning tests.

The article includes general-arrangement and midship-section drawings, machinery-layout drawings, and photographs of the bridge and control-room consoles.

26,713 ***Balbina* Japanese-Built Multi-Purpose Carrier.** *Shipp. World & Shipb.*, **160** (1967), p. 1727 (Oct.) [6 pp., 2 tab., 3 graphs, 1 diag., 5 phot.]

The 75,670-ton d.w. oil ore bulk carrier *Balbina* was built by the Kure Shipbuilding & Engineering Co. Ltd. for S.A. di Navigazione Marittima Dorado, of Switzerland, and ran sea trials in June 1967. Her principal particulars are:

Length, o.a.	254·5 m (834·75 ft)
b.p.	243 m (797·2 ft)
Breadth, moulded	36·5 m (119·75 ft)
Depth, moulded	20 m (65·6 ft)
Design draught, moulded	12·9 m (42·3 ft)
Maximum draught, moulded	13·4 m (43·9 ft)
Gross tonnage, Liberian	43,449·8
Deadweight at design draught	75,673 tons
Deadweight at maximum draught	79,450 tons
Capacity (100% full)	
Cargo holds, grain	96,242 cu m (3,398,800 cu ft)
Ballast water	31,081 cu m (1,097,600 cu ft)
Fuel-oil tanks	4,227 cu m (149,275 cu ft)
Trial speed on 40-ft draught	16·38 knots
Service speed on 42·5-ft draught	15 knots

The ship has a single continuous freeboard deck, a short raised forecastle, a six-tier superstructure aft (the four upper tiers being separated from the funnel), a bulbous bow well faired into a soft-nosed clipper stem, a transom stern, and a " Mariner " type stern frame and rudder. There are nine cargo holds, Nos 4 and 6 being shorter than the rest. Each hold is served by a single hatch fitted with MacGregor side-rolling covers. All the holds can be used for grain, ore, grade "A" petroleum products, or water ballast. Shifting boards are not required. On each side of the cargo section there are five upper wing tanks which can also be used for water ballast. The double-bottom tanks in the cargo section can carry water ballast and or fuel as required. Fuel is also carried in deep tanks forward of No. 1 hold and in two deep wing-tanks in way of the forward end of the engine room.

Heavy cargoes can be carried in alternate holds with the remainder empty. The hull, including the double bottom in way of the holds, is longitudinally framed, and the holds are separated by vertically corrugated bulkheads; other bulkheads are of flat plate with vertical stiffeners. The double bottom in the engine room is of cellular construction. There are two bilge wells (one port and one starboard) at the after end of each cargo hold, and port and starboard pipe tunnels run the full length of the cargo section. A double main-ballast line runs through a duct keel forward from the pump room; the lines are of 450 mm (17·7 in) diameter, and a hydraulically-operated valve and a special bell-mouth are fitted on each suction. Stripping is performed by a 150-tons/hr water eductor which is supplied from the ballast pump.

There are two steam-turbine driven main cargo pumps, each of 3,000 tons/hr capacity, and two 300 tons/hr steam duplex pumps. Two different grades of oil can be handled simultaneously by means of two independent oil-line systems, one for holds Nos 1 to 4 and the other for holds Nos 5 to 9. The Golar Vent-Dry system for gas-freeing and drying the holds is described; it consists of a steam-turbine driven fan and air-heating steam coils. Oil cargo can be heated by steam coils from 41 F to 150 F when the sea temperature is 41 F. Cargo valves can be operated hydraulically from a control station at the front of the deck-house, or from the pump room. Valve positions and liquid levels are indicated in the control station.

The main engine is an I.H.I.-Sulzer 8RD90, with a normal output of 16,500 b.h.p. at 118 r.p.m. and a maximum continuous rating of 18,400 b.h.p. at 122 r.p.m. It drives a 20·6-ft diameter five-bladed propeller, which can be removed without unshipping the rudder.

Electrical power is obtained from one 600-kW turbo-alternator having a multi-stage waste-heat turbine, and two 975-b.h.p., 650-kW Diesel sets. There is also a 100-kW Diesel emergency set. Steam-raising plant consists of a two-drum, single-burner oil-fired boiler rated at 47 tons/hr at 227·6 lb/sq in and 446 F, and a La Mont type exhaust-gas boiler rated at 5·5 tons/hr.

Details of some fuel-consumption calculations relating to the ship's official sea trials when running under (*a*) maximum, and (*b*) normal power conditions are given. Analyses of the Diesel oil and the bunker oil used during (*a*) and (*b*) respectively are shown in a table. Speed/power curves obtained during sea trials are given. A Geiger torsiograph was used to measure the additional stresses due to torsional vibration in the crankshaft and propulsion shafting in the speed range 25 to 140 r.p.m.; results are presented in tabular and graphical form.

General-arrangement drawings of the ship, and photographs of the engine room, the main control console, and the deck layout are included.

26,714 **Two 20,700-ton d.w. Bulk Carriers Join the Sugar Line Fleet.** *Motor Ship*, **49** (1968), p. 161 (July) [7 pp., 2 tab., 8 diag., 8 phot.]

The bulk-sugar carriers *Sugar Crystal* and *Sugar Producer*, both built by the Port Glasgow yard of the Scott-Lithgow group, recently entered the service of Sugar Line Ltd, a company in the Tate & Lyle group. These sister ships carry unrefined sugar from the West Indies to London; as this is a seasonal trade, they had also to be suitable for other bulk cargoes. Further requirements which had to be met included ship dimensions to suit the discharging terminal at the refinery at Silvertown, on the Thames. The unloading wharf (which has a conveyor system) has been increased in size, allowing the ships to be longer than the Line's older vessels (e.g. the 8,500-ton d.w. *Sugar Carrier*, built in 1960—see Abstract No. 16,616, July 1960). The length chosen was 550 ft overall; this is almost the maximum length if the ships are to be able to turn in the river near the terminal. There were also draught limitations to consider, particularly at this terminal, and breadth was restricted to enable the ships to use the St Lawrence Seaway. The ships can carry 17,500 tons of sugar on a f.w. draught of 29 ft. The maximum draught of 31 ft 4 in will be usable for ore cargoes and for grain with a stowage factor heavier than 50 cu ft/ton; this draught is obtained on a "B-60%" freeboard, under the 1966 International Convention on Load Lines (the *Sugar Crystal* was the first British-built ship to take advantage of these new regulations). The principal particulars of the two ships are: –

Length, o.a.	550 ft
b.p.	520 ft
Breadth, moulded	73 ft
Depth, moulded	41·5 ft
Draught, loaded	31·3 ft
Deadweight, corresponding	20,700 tons
Draught, light	8·1 ft
Light ship weight	6,155 tons
Register tonnage	13,894 gross, 8,203 net
Block coefficient, on load draught	0·789
Service speed	15½ knots
Complement	32
Cargo capacity, hold (grain)	677,173 cu ft
hatches	54,184 cu ft
wing tanks	309,244 cu ft
Total	1,040,601 cu ft

The description given in the article applies to both ships, and includes general-arrangement drawings, a midship-section drawing (giving scantlings and indicating scantling reductions made possible by applying "corrosion control"), and machinery-arrangement drawings.

The engine room and superstructure are aft; there is a short forecastle and a poop. The five holds are designed to facilitate the handling of sugar by the owners' 12½-ton shore cranes, which are fitted with grabs. The longitudinal bulkheads are 5 ft outboard of the hatch square, thus reducing the likelihood of damage from the grabs. Nos 2 to 5 holds are flanked by wing tanks extending from top to bottom; No. 1 hold has upper wing ballast-tanks and combined double-bottom hopper tanks. Grain can be carried in the holds and side tanks. The article includes a table of tank capacities, and a deadweight scale.

For handling cargoes other than sugar, each ship has ten 7½-ton derricks; there is one pair at the forward end of No. 1 hold, and two pairs are mounted between Nos 2 and 3 holds and two pairs between Nos 4 and 5 holds. Each boom is stowed in a raised position athwartships, diagonally from the foot of its post to the head of the other post of the pair. For periods of trading operations during which the ship's cargo-handling equipment is not used, masthead platforms provide access for fitting and removing the derrick blocks (while the booms are in the stowed position). This stowage arrangement also allows the hatches to be opened without the need to top the derricks.

The ten cargo winches, all of 7½ tons capacity and supplied by the Norwinch group, are hydraulically driven; those serving Nos 1 and 5 holds are self-tensioning for mooring and Seaway purposes. There is also a hydraulic 12-ton self-tensioning mooring winch aft, and a hydraulic windlass forward. The cargo winches (apart from the two on the forecastle deck) are mounted on mast-houses.

A Lithgow ram bow is fitted. The stern frame is of the clearwater type. The spade rudder, which is comparatively small and of Burmeister & Wain design, can be readily inspected and its bearings can be changed without difficulty (there is a risk of the rudder bottoming at the sugar terminals). The article includes drawings of the stern, showing the rudder arrangement. (See also Abstract No. 23,856, Jan. 1966.)

The main deck has a ⅛-in coating of Bargex (a composition produced by Universal Highways); the deckhouses and the hold surfaces are coated with epoxy paints. Other anti-corrosion measures include non-metallic partitions in the accommodation shower-rooms, some plastics-lined stripping lines, and Tufnol bushes in the fairleads and in the grain-hatch toggle hinges (these toggles, shown in a photograph, can be tightened by a single turn of the spanner).

The two ships are equipped with the first examples of Marconi Marine's integrated bridge console; the console, which includes the steering unit, is set 15 ft back from the sloping windows of the wheelhouse front (certain navigation and other instruments are mounted above the windows). The article briefly describes the layout of the console equipment.

The main engine, a Kincaid B. & W. 674 VT2BF-160 rated at 9,000 b.h.p. at 115 r.p.m., is controlled from a well-equipped console on the engine-room lower flat. Ballast operations are controlled from a separate panel in the engine room by a Hansen hydraulic system.

There is accommodation for a complement of 33. As in the owners' earlier ships, there is a main double stairway. Store rooms are centralised, enabling a portable roller conveyor to be used.

26,715 ***Frederick Carter*** - **Twin-Screw Train Ferry for Newfoundland Service.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 822 (14 June) [3 pp., 1 tab., 8 diag., 2 phot.]

The *Frederick Carter* is a large icebreaking rail and vehicle ferry built for Canadian National Railways by Davie Shipbuilding Ltd, Lauzon, Quebec, for operation between Nova Scotia and Newfoundland. Her principal particulars are:

Length, o.a.	487 ft
b.p.	450 ft
Beam, moulded	69 ft
Depth, moulded, to main deck	27 ft
upper deck	48 ft
Draught, scantling	21 ft
summer, salt water	20 ft
Service speed	18 knots
Summer displacement (approx.)	10,875 tons
Crew	66
Capacity	
Road trucks	12
Freight cars (rail)	39

The transversely-framed hull is built to Lloyd's Register Class 1 for operation in ice, and has a bow raked aft below the waterline to assist icebreaking. The heeling tanks which, together with trimming tanks, have been fitted to control heel and trim while freight cars are being loaded and unloaded, can also be used to assist in freeing the vessel from an ice field by a rapid transfer of water from one side to the other.

The machinery is amidships, and exhausts through side trunks and funnels. There are four 12-cylinder PC2V400 Crossley Pielstick engines, two geared to each of the two propeller shafts driving KaMeWa controllable-pitch propellers of 13 ft 5½ in diameter through 2·6 to 1 De Schelde gearboxes and Vulcan Sinclair hydraulic couplings. The engines, which have a m.c.r. of 6,000 b.h.p., develop 3,718 b.h.p. each at 483 r.p.m.

The ship can carry 39 loaded rail freight cars, each 45 ft long, on five lines of track at main-deck level, and 12 large road trucks at upper-deck level. Loading and discharge of the freight cars takes place through the after end of the totally-enclosed tweendeck; the road trucks are driven on and off at upper-deck level by raised shore ramps. A Flume-type passive roll-stabilising system has been fitted; it incorporates a fast-acting gravity dump system. A Norris Warming high-velocity ventilation system copes with the exhausts of Diesel-driven compressors on refrigerated vehicles.

The jet type bow thruster is specially designed to work effectively in broken or slush ice, the pump inlet being placed well below the waterline. The streamlined rudder is also of special design and strengthened for navigation in ice. An ice knife under the stern, in way of the rudder stock, protects the rudder when the vessel backs into ice.

The wheelhouse is totally enclosed to give maximum protection during the year-round operation, much of it in severe weather. For the same reason, special attention has been given to heating, ventilating, and insulating the crew accommodation.

26,716 ***Europic Ferry* A New Ro-Ro Ship for the Felixstowe Europort Route.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 232 (16 Feb.) [5 pp., 1 tab., 2 diag., 11 phot., 2 diag.]

The *Europic Ferry* was built by Swan Hunter for the Atlantic Steam Navigation Company's Felixstowe Europort service and provides a fast road sea cargo service between the U.K. and the Continent. Her principal particulars are:

Length, o.a.	451·3 ft
b.p.	426 ft
Breadth, moulded	66·5 ft
Depth, moulded	38 ft
Draught, maximum	15 ft
Deadweight	3,020 tons
Gross register	4,770 tons
Propulsive power (m.c.r.)	2 × 6,800 b.h.p.
Service speed (85% m.c.r.)	18 knots
Vehicle capacity	
large commercial vehicles	120

The ship was built to Lloyd's highest requirements for her type, and, in accordance with the owners' stipulation, the shell is riveted to the frames. The engine room is placed well forward and is over-run by the main vehicle deck, into which only a small machinery casing protrudes. Aft of the engine room, below this main deck, there is a small cargo space with ramp access from the main deck. The only other through deck is the upper deck, which (the superstructure being well forward) provides further motor-vehicle and cargo-stowage space. The portable intermediate wing car decks generally carried by roll-on/roll-off vessels are absent, because the owners intend to concentrate on year-round commercial vehicle traffic rather than seasonal private cars. Vehicles enter and leave by means of shore ramps to the upper deck or by the ship's own drawbridge-type stern ramp to the main deck. This ramp, and a 78 × 12 ft ramp linking the two decks, are hydraulically operated. The enclosed vehicle spaces are ventilated by eight flameproof reversible axial-flow fans, four aft and four forward.

The propulsion machinery is designed to run on fuel of viscosity 1,000 sec Redwood at 200–210° F. It consists of two 16-cylinder turbocharged reversible Lindholmen Pielstick engines, each developing 6,800 b.h.p. at 450 r.p.m. and driving a Stone KaMeWa controllable-pitch propeller through Twiflex couplings and Hindmarch MWD R.10 9 : 5 reduction gear. The shaft system is over 230 ft long, which is unusual for a ship of this size; only the tailshafts are hollow. The engines can be controlled from the engine room, the wheelhouse, and bridge wings; but the system is not highly automated since it is not intended to operate with an unmanned engine room on this comparatively short route with frequent manœuvring. Extensive navigational aids are, however, carried because the route cuts across major traffic lanes and is subject to bad weather conditions. Decca HD516 and TM626 radar are installed, together with the Decca Navigator. An 800-h.p. KaMeWa bow-thrust unit assists docking without tugs.

The accommodation, which is of unusually high standard, provides two- and four-berth cabins for 44 passengers. It is air-conditioned throughout by a Norris electric reheat system, which enables the temperature of each cabin to be adjusted to the wish of the occupants without the need for dual ducts. The preheated air enters each cabin through an electric re-heat attenuator containing an electric element controlled by a variable thermostat in the cabin. (See also Abstract No. 25,094, Feb. 1967.)

General-arrangement drawings are given. At the end of the main article there is a brief description, with two diagrams, of the procedure used for "fair-curve" alignment of the transmission shafting.

26,717 ***Humboldt*** **A Versatile Automated Gas Carrier.** *Shipbuild. Shipp. Rec.*, **112** (1968), p. 46 (12 July) [4 pp., 1 tab., 2 diag., 7 phot.] **The *Humboldt* —An Automated L.P.G. Tanker.** *Motor Ship*, **49** (1968), p. 115 (June) [4½ pp., 3 tab., 8 diag., 6 phot.]

The gas carrier *Humboldt* is suitable for the carriage of gas at low temperature or at high pressure. She can load a cargo totally or partly refrigerated, and also non-refrigerated products which can be refrigerated on board. She was built by Chantiers Navals de la Ciotat to the order of Ocean Gas Transport Ltd (part of the Houlder group), and classified by Bureau Veritas ✠ 3/3 L.I.I.A. & C.P. L.P.G. Carrier. Her principal particulars are

Length, o.a.	116·95 m (383·7 ft)
b.p.	105·70 m (346·8 ft)
Breadth, moulded	16·50 m (54·1 ft)
Depth, moulded	8·80 m (28·9 ft)
Draught, summer freeboard	6·50 m (21·3 ft)

Deadweight	5,165 tons
Displacement	8,327 tons
Cargo-tank capacity	6,250 cu m (220,720 cu ft)
Propulsive power	5,600 b.h.p. at 225 r.p.m.
Service speed	15 knots

An automated engine-control system with bridge control has been installed which enables the engine room to operate without regular watchkeepers, and to be unmanned at night (between 17.00 and 07.00 hrs) and during weekends. In addition, the deck machinery and mooring equipment have been so arranged as to ensure high reliability and simplification of handling operations. As a result of these measures, agreement has been reached with the National Union of Seamen for the *Humboldt* to operate with an experimental " semi-general-purpose " crew, members of which are allowed to undertake duties on deck or in the engine room. All receive a productivity bonus, but the reduction of the total complement to 23, which includes only six engineers in all, has resulted in improved economy of operation.

The vessel can carry ammonia, propane, propylene, butane, or butadiene, at between 48 C (54 F) at atmospheric pressure and the temperature corresponding to 6·3 atmospheres, in six horizontal cylindrical cargo tanks, two partly above the main deck and four below decks. Their total capacity is 6,250 cu m (220,720 cu ft), and they are designed to withstand a vacuum of 80%, as well as the maximum working pressure of 6·3 atmospheres. They are made of high-grade steel conforming to Technigaz specification TGZ201, grade 2, and are insulated with sprayed-on rigid polyurethane foam, 60-mm (2·4 in) thick on those parts above deck and 80-mm (3·15 in) on those below. The cargo lines have been insulated with the same material injected between the pipes and stainless-steel cladding.

Cargoes are refrigerated by four plants, each comprising a Loire compressor, a condenser, and a droplet separator. The installation is designed to cool propane or ammonia from 35 C (95 F) to between 5 C and 5 C (41 and 23 F), and to maintain it at this temperature. Propylene can be maintained at 48 C (54 F), propane at 42 C (44 F), and ammonia shipped at atmospheric pressure at 33 C (28 F). For discharging cargo there are three electrically-driven Worthington pumps, each with hydraulic coupling and speed-increasing gear. The motor speed of 3,800 r.p.m., at which the flow rate is 180 cu m (6,357 cu ft)/hr against a head of 282 m (922 ft), can be reduced to 2,740 r.p.m. for the same flow rate against a 140-m (460 ft) head.

The main engine is a seven-cylinder M.A.N. type K7Z 57/80E Diesel with two Brown Boveri turbochargers and water-cooled pistons. It is equipped to operate on residual fuel, develops 5,600 b.h.p. at 225 r.p.m., and drives a four-bladed Lips propeller of 3·5 m (11·5 ft) diameter. It is overlooked by a centralised control room in which there is a 110-point automatic scanning and alarm system (but no print-out facility). Some deviations in pressures and temperatures will slow the engine; others will stop it. All main-engine pumps have automatic cut-in of the standby units. The auxiliaries comprise three M.A.N. G5V 23·5/33 mAL Diesels, each developing 500 h.p. and driving a Siemens 320-kW alternator. These Diesel-alternators are not self-starting but have automatic selective cut-out arrangements.

The alarm system is connected to the second, third, fourth, or fifth engineers' cabins, selected according to duty schedules. Audible alarms are given on the bridge, and in the control room where they are all dealt with. An Icare installation detects dangerous gas concentrations.

Both articles include general-arrangement drawings and a midship section; *Motor Ship* also gives machinery-layout drawings and a deadweight scale.

26,718 ***Esso Baltica* A Distinctive Products Carrier.** *Motor Ship*, **49** (1968), p. 129 (June) [5 pp., 1 tab., 1 diag., 14 phot.]

The *Esso Baltica* is a 5,533-ton d.w. products carrier which can carry up to four types of oil products and discharge any three simultaneously. Her engine room has been designed for operation without watchkeepers, and to be unmanned at night and during weekends. Except for a harbour/emergency set, there are no auxiliary generators; instead, two 1,400-kVA alternators are driven off the main gearbox. This high generator output is required for the operation in port of the three deep-well cargo pumps and a 300-h.p. bow thruster.

The principal particulars are:

Length, o.a.	360·6 ft (109·9 m)
b.p.	334·6 ft (102 m)
Breadth, moulded	53·0 ft (16·15 m)
Depth to main deck	24·9 ft (7·60 m)
Draught, summer freeboard	20·1 ft (6·12 m)
Deadweight	5,533 tons (5,622 metric tons)
Register, gross	3,708 tons
net	1,969 tons
Number of cargo tanks	2 × 5
Cargo-tank capacity	7,512 cu m (47,300 barrels)
Speed, loaded	13 knots
Propulsive power	2 × 1,910 b.h.p.
Total complement	19

The ship was built in Holland for Dansk Esso by A. Vuyk en Zonen's Scheepswerven N.V. General-arrangement drawings are given.

Most of the article is devoted to a description of the main and auxiliary machinery. The two main engines, each developing 1,910 b.h.p. at 500 r.p.m., are M.A.N. type G9V 30/45 ATL non-reversing turbocharged four-stroke trunk-piston Diesels. They drive a controllable-pitch KaMeWa propeller, 3·75 m (12·3 ft) in diameter, at a constant speed of 170 r.p.m. through Vulcan hydraulic couplings and an AG Weser reduction gearbox. Each engine has an extended shaft which passes through a hollow shaft in the gearbox and drives a constant-speed Thrige-Titan autovolt 1,400-kVA three-phase 60-cycle 440-V alternator at 1,200 r.p.m. through a speed-increasing gear. At sea, with both engines running, the two alternators, coupled in parallel, will cover the normal electrical load; in port, while cargo is being discharged, one Diesel with its alternator will meet all the electrical power requirements, including those for lighting, all cargo pumps, and hydraulic deck machinery. While cargo is being loaded, both engines are stopped, and electric power is supplied by a 265-h.p. 230-kVA Diesel-alternator set.

As the engine-room staff have no watchkeeping duties, they have more time for maintenance. One engine can be overhauled while the other is in use for propulsion or cargo pumping. With one engine, a speed of 11 knots can be maintained. All manœuvring is effected from the bridge, by pitch adjustment at constant shaft speed. The Lips bow thruster is hydraulically driven, and controlled from the bridge.

There are two Worthington deep-well cargo pumps, each with a capacity of 750 cu m/hr (26,480 cu ft/hr) of sea water at a head of 100 m (328 ft), and a deep-well ballast cargo pump of capacity 385 cu m/hr (13,590 cu ft/hr), also at 100-m head. To save space in the engine room, these three pumps are driven by electric motors in a compartment in the forward part of the superstructure, the drives passing through gas-tight glands in the superstructure wall.

To reduce maintenance and obtain a reduction in scantlings allowed by the Classification Society, all ten cargo tanks, the main deck, the forecastle and poop decks, the tank lids, and the cargo pipes and ladders inside the tanks have been coated with Esso Rustban 191 anti-corrosive, which is expected to be effective for several years.

The combination of constant-speed engines and c.p. propeller, and the absence of auxiliary engines, has simplified the arrangement of automatic controls required for the periodically unmanned engine room. The glass-enclosed control room in the engine room is neither air-conditioned nor sound-insulated. It contains an alarm panel for 40 points scanned by a Lyngso system, and there are arrangements for the automatic cut-in of certain auxiliaries such as the stand-by hydraulic motor for pitch control of the KaMeWa propeller, lubricating-oil pumps for the reduction gear and the main engines, and other important services.

Steam for cargo-heating and other purposes is supplied by three Clayton steam generators producing 2½ tons of steam per hour each. For fire protection, the new GW Sprinkler A/S "dry foam" system is installed. This enables the engine room to be filled completely with light-expansion foam in four minutes. The foam contains very little water and is therefore said not to harm electrical equipment. It also contains enough air to enable personnel to remain safely in the engine room.

Although the number of crew required to operate the ship is smaller than in a conventional vessel of comparable size, no interchange of duties between deck and engine room is required.

26,719 ***Orwell Fisher*** **Container Ship with Cellular Provision.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 370 (15 Mar.) [2 pp., 1 tab., 2 diag., 3 phot.]

The container ship *Orwell Fisher*, delivered to James Fisher & Sons (of Barrow) early in 1968, was built by Gebr. van der Werf (of Deest, Nijmegen). A second ship, the *Solway Fisher*, was scheduled for delivery from the same builders in May 1968. These sister ships are for long-term charter to Atlantic Steam Navigation for service between Preston and Ireland. Their principal particulars are:

Length, o.a.	90·1 m (295·6 ft)
b.p.	83·5 m (274 ft)
Breadth, extreme	15·21 m (49·9 ft)
moulded	15 m (49·2 ft)
Depth, to shelter deck	7·6 m (24·9 ft)
to tweendeck	4·65 m (15·3 ft)

Draught	4·63 m (15·2 ft)
Deadweight	2,465 tons
Gross register	1,500 tons
Service speed	14½ knots
Classification	LR ✠ 100 A1, LMC

Each ship can take Lancashire flats as well as containers. This " free stowage " cargo arrangement may be modified later to provide a cellular arrangement for ISO containers; this would allow the maximum number of 20-ft containers to be increased, from the present 89, to 109 or 116 (depending on whether those on the upper deck were stowed four or five abreast). The hull is designed so that such modification can readily be done; the double bottom has been reinforced for cellular construction. Special container-notches are fitted on the tank top and on all the hatch covers.

The all-aft layout of the ship is shown in general-arrangement drawings. These include cross-sections showing the ship (*a*) as an open shelterdecker (the present arrangement) with one tier of containers on the tank top, one on the main deck, and one on the upper deck, and (*b*) as a closed shelter-decker (the cellular arrangement) with three tiers of containers supported by the tank top and one tier on the upper deck. The hatches are very large; the forward hatch on the upper deck is 14·72 m long by 11·13 m (48·3 by 36·5 ft), and the after one is 32·4 m long by 11·75 m (106·3 by 38·5 ft); the main-deck hatches are slightly smaller. The hatch covers, of the MacGregor hydraulic type, can all be opened within 4 min. There is no cargo-handling gear.

Accommodation is provided for a complement of 19. A Van Ommeren stabiliser tank (see Abstract No. 24,411, July 1966) is arranged on the tank top between frames 85 and 90 (i.e. about 1/3 L aft of FP).

The main engine, which can be controlled from the bridge, is a geared eight-cylinder Deutz RBV 8M358 Diesel developing 2,450 b.h.p. at 350 r.p.m. There are three 90-kVA Diesel-alternator sets.

26,720 **Drill Rigs.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 237 (16 Feb.) [2 pp., 1 tab., 3 diag.]

This article is divided into two sections. The first describes a Dutch design for a semi-submersible drilling platform, and the second an invention for improving the stability of a drill rig.

Dutch Semi-Submersible Drilling Platform. This rig, called *Norrig-5*, will remain afloat even if three of its five columns are missing.

Tests in the Wageningen Model Basin have demonstrated its satisfactory behaviour in a seaway, and the designers, Ingenieursbureau Marcon NV, estimate that its number of weather working days will be 5–10% higher than that of other rigs taking comparable working loads—some 2,650 tons of equipment. It has been designed to withstand extreme conditions of 60 ft waves, a current of 6 knots, and a sustained wind of 115 m.p.h. gusting to 140 m.p.h. A cut-away sketch is given. The main particulars are:

Length, o.a.	87·80 m (288 ft)
Breadth, o.a.	91·45 m (300 ft)
Depth, to upper deck	57·00 m (187 ft)
to lower deck	50·90 m (167 ft)
Draught, in floating drilling condition	29·88 m (98 ft)

Displacement, corresponding	25,300 tons
Draught in fixed legs-on-bottom condition	35·97 m (118 ft)
Displacement, corresponding	28,000 tons
Draught, towing condition	9·15 m (30 ft)
Displacement, corresponding	12,000 tons

Improved Drill Rig Stability. The principle described is due to Captain B. Peri, of Yeda Yami Ltd, Haifa. Its essence is that the extracting forces that occur during drilling operations are applied to the rig at a clamp which is below the c.g. of the structure. Hence the greater the extracting force the more stable the rig becomes, whereas the opposite is true for a conventional rig. Other advantages follow from the increased stability and smaller relative size of rigs constructed on this principle. They include simplified anchoring and position keeping; the possibility of exerting high extracting forces without assistance from another craft; greater safety for the operating crew; elimination of towing to the operating site, because the components can be shipped as cargo and assembled afloat; and increased depth of drilling.

26,721 ***Bärbel Bolten* Container Vessel for Short Sea Trades.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 445 (29 Mar.) [3½ pp., 2 tab., 5 diag., 8 phot.]

The *Bärbel Bolten* was built by Schlichting-Werft, Travemünde, for Reederei Aug. Bolten, and will be used on the liner service between Ireland and England. The owners required a vertical-loading specialised ship, with no cargo-handling gear, able to stow between 80 and 90 containers either of the international type (20 ft × 8 ft × 8 ft) or of British Rail type (20 ft × 8·3 ft × 9 ft) or both. The speed was to be about 14½ knots, the gross register not to exceed 999 tons, and the draught fully-loaded not to exceed 3·82 m (12 ft 6¼ in). The vessel as built is of all-aft layout and has the following principal particulars:

Length, o.a.	86·80 m (284·8 ft)
b.p.	79·00 m (259·2 ft)
Breadth, moulded	14·60 m (47·9 ft)
Depth, moulded	7·55 m (24·8 ft)
Deadweight	2,315 tonnes (2,279 tons)
Capacity, grain	5,380 cu m (190,000 cu ft)
bale	4,950 cu m (174,800 cu ft)
Propulsive power	2,500 b.h.p. at 325 r.p.m.
Speed	14½ knots
Radius	6 000 miles

The ship was built to Germanischer Lloyd Class ✠ 100 A4 EI ✠ MC EI. The depth of the double bottom, which is of longitudinal-girder construction, varies along the length. In the forepart of the hold it is the maximum depth permitted by measurement rules; further aft it permits international containers to be stowed in the lower hold, and still further aft British Rail containers. Twenty-one containers can be carried in the lower hold, 31 in the tweendecks, and 36 on deck. Over the entire length of the hold there is no area out of hatchway. On the main deck the hatch width (11·90 m 39 ft) extends over 81% of the breadth, and the total length of hatchway (51·20 m 168 ft) is about 65% of the length b.p. The extension of the full width of hatch as far forward as possible has led to marked flare in the bow section.

The vessel has a Maierform SV bulbous bow (see Abstract No. 25,044, Feb. 1967) which, by model-basin tests, was shown to result in a reduction of 13 % in required power and also to improve sea behaviour considerably. This has been confirmed by the sea trials of the *Bärbel Bolten*. To reduce further the loss of speed in a seaway, Flume passive stabilising tanks are fitted. The reduced rolling is also of importance in view of the stowage of containers on deck. Moreover, the two Flume tanks substantially increase the torsional rigidity of the hull.

The large folding hatch covers are operated hydraulically; all main-deck hatches can be opened in about 8 min. Flush non-watertight hatches have been fitted in the tweendeck.

The machinery installation has been planned for a 16-hour unmanned period for the engine room, and special attention has been paid to reliability, including the use of vibration-free equipment and resilient mountings, and the prevention of tube and pipe fractures due to hull vibration. The main engine is a six-cylinder MaK type 6 Mu 551 Ak Diesel developing 2,500 b.h.p. at 325 r.p.m., and directly coupled to the 2·40 m (7·9 ft) dia. propeller.

26,722 ***Planet* Self-Loading Discharging Bulk Cement Carrier for Danish Owners.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 334 (8 Mar.) [2 pp., 2 tab., 1 diag., 4 phot.]

The bulk cement carrier *Planet*, designed by Martin A. Nielsen (the Copenhagen consulting engineers), was recently completed by the Aarhus Flydedok & Maskinkompagni for the AB Aalborg Portland-Cement-Fabrik service between North Jutland and storage installations in Copenhagen and elsewhere in Denmark. A sister ship, the *Kongsdal*, was built by Aarhus for the same owners in 1965. The principal particulars of the *Planet* are:

Length, o.a.	79·8 m (261·8 ft)
b.p.	73·3 m (240·5 ft)
Breadth, moulded	12·9 m (42·3 ft)
Depth, moulded	6 m (19·7 ft)
Draught	4·8 m (15·7 ft)
Deadweight	2,120 tons
Capacity	1,550 cu m (54,740 cu ft)
Register tonnage	1,535 gross, 629 net
Service speed	12 knots

The ship is of all-aft layout. She has a forecastle, poop, raked stem, and cruiser stern. Ice-strengthening is incorporated. Five transverse bulkheads divide the hull into the following main compartments: fore peak and chain locker; forward deep tank and bow thrust-unit space; forward hold; after hold; engine room; and after peak. Wing tanks, for ballast and fresh water, are formed between longitudinal bulkheads and the ship sides. A centreline bulkhead divides each of the two holds into two compartments. The hull is cathodically protected; the holds were left unpainted.

For loading, the cement is transferred to the ship through a flexible hose connected to a screw conveyor mounted transversely on a deckhouse (this is amidships, above the elevator room). The cement is transferred from this transverse conveyor to four longitudinal screw-conveyors, from which it is dropped through deck apertures into the four cargo compartments. Air is sucked out from the holds by electric ventilator units provided with filters. Loading time is 2 to 2½ hr. for a full load of 2,000 tons.

For discharging, two chain conveyors running in V-channels along the bottom of each of the four cargo compartments carry the cement to the elevator room (located between the two holds), where vertical screw-conveyors lift it to the upper deck. From here, it is taken over the ship's side by the transverse conveyor. Discharging takes 6 to 8 hr.

Loading and discharging are fully automatic, and are controlled from a panel in the deckhouse. The propeller pitch and r.p.m., and the KaMeWa bow thruster, have bridge control. The accommodation for the crew of 20 is in the superstructure and poop. All members of the crew have single cabins.

The main engine is a 12-cylinder B. & W. Diesel, type 1226 MTBF-40V, developing 1,980 b.h.p. at 600 r.p.m. It drives a KaMeWa c.p. propeller through 1 : 2·5 Renk reduction gearing. There are three 145-kVA alternators, each driven by a Scania Vabis 1,500-r.p.m. Diesel.

26,723 **Passenger and Car Ferry for Service in the Netherlands.** *Holland Shipbuild.*, **16** (1967), p. 44 (July) [3½ pp., 1 tab., 2 diag., 7 phot.]

The *Prins Willem IV* is a twin-screw shallow-draught passenger and car ferry for service on the Waddenzee; she runs between Holwerd on the Frisian mainland and the island of Ameland. The ship is owned by the Leeuwarden Rijkswaterstaat (i.e. Roads and Waterways Administration); she was designed by Scheepswerf Barkmeijer, Vierverlaten, and built by Scheepswerf Bijlholt, Foxhol. Her principal particulars are:

Length, o.a.	50 m (164 ft)
b.p.	47·3 m (155·5 ft)
Breadth, o.a.	9·4 m (31 ft)
moulded	8·8 m (29 ft)
Depth, moulded	2·6 m (8·5 ft)
Draught, loaded	1·1 m (3·6 ft)
Car capacity	32
Passenger capacity (maximum)	390
Seating capacity	316

In profile the ship resembles an oil-rig supply vessel. There is a long open car-deck having straight sheer, and a flat transom stern with radiused corners. Immediately aft of a short raised forecastle deck is a superstructure which spans the full width of the forward end of the car deck. The top of the superstructure forms an open bridge-deck and carries, well forward, an enclosed wheelhouse with all-round vision. Aft of the wheelhouse there are seats for 92 passengers in the open. Below the bridge deck is an enclosed deck-saloon for 121 passengers and cafeteria facilities; aft of this saloon, and on the same level, are a further 25 seats in the open. Toilet facilities, stores, and a mail room are arranged forward in the superstructure at car-deck level. Below the forward end of the car deck is a second saloon seating 96 passengers; this is divided over half its length by a centre-line alleyway providing access to the engine room from the superstructure without obstructing the car deck. Aft of this saloon, and separated by the alleyway, are two wing ballast tanks, a cabin for two crew members to port, and a heating-plant compartment to starboard. Special attention has been paid to ventilation below the car deck.

The engine room extends from amidships to about three-quarters aft, where it has access to a large compartment housing a Seffle electro-hydraulic steering gear with double-acting hydraulic rams for the two balanced rudders. In way of this compartment are additional wing ballast tanks.

The car deck, which has a slight camber, can accommodate 32 medium-sized cars. Access to this deck is over hydraulically-operated hinged ramps made by Clausen KG, of Oberwinter; they are arranged in seven panels on each side, with a total length of 59 ft. When closed, the panels form a deep protective bulwark in continuation of the superstructure sides. The after part of the car deck has a guard rail.

Propulsion is by two Kromhout 12-cylinder Diesel engines, each developing 350 b.h.p. at 1,800 r.p.m. They drive 3·75-ft diameter three-bladed manganese-bronze propellers at 500 r.p.m. through Brevo hydraulic reverse reduction gears. Auxiliary machinery is arranged in two groups. The first group comprises a 42-h.p. Diesel engine which drives a 23-kVA alternator, a 4-in pump of 3,530 cu ft/hr capacity, an air compressor, and a fire pump. The second group comprises a 165-h.p. Diesel engine which drives a 10-kW emergency generator, an auxiliary compressor, and a D.C. generator feeding the 125-h.p. two-speed electric motor for the Naviprop bow-thrust unit. All machinery is resiliently mounted.

The operation of the bow-thrust unit is described. Its horizontal propeller draws water from below through large openings in the shell and pumps it to port or starboard via electro-hydraulically operated valves in the ship's sides. A thrust of 2,870 lb is available within six seconds, and its direction can be reversed in two seconds. This installation was supplied by Clausen, and is the first of its type to be fitted in a Netherlands vessel.

The article includes general-arrangement drawings, photographs of the ship on trials, and a list of some of the equipment suppliers.

INDUSTRIAL AND ECONOMIC INFORMATION

26,724 **The Economics of Automation in British Shipping.** Goss, R. O. *Trans. R.I.N.A.*, **109** (1967), p. 347 (July) [14 pp., 7 ref., 16 tab.; and Discussion: 2 pp., 1 ref., 1 tab.]

This paper discusses the purely economic aspects of automation or other increases of capital-intensity in British ships and, in particular, short-cut methods which may be employed in deciding whether the fitting of any given piece of equipment will be profitable. Because such decisions may involve peculiarly difficult (complex and uncertain) financial estimates, the suggested methods concentrate upon determining, first, the minimum profitability necessary to justify a known capital cost, and, second, the maximum capital cost justified by a known increase in profitability. Use is made of the Net Present Value (NPV) technique adopted (for comparing ship designs) in the Author's earlier paper covered by Abstract No

24,475 (Aug. 1966). Investment grants, corporation tax, and free depreciation for tax purposes are combined into two alternative tax positions for the shipowner, and account is taken of constant and of geometrically increasing effects on the ship's profitability. The results of the preliminary stages are tabulated, and the final results are presented in the form of ready-reckoner tables.

An appendix contains a revised version of the calculation, given in Appendix II of the earlier paper, of the NPV of a hypothetical ship project. The calculation has been brought up to date in terms of the new system of tax, etc., and employs many of the preliminary results shown in the body of the paper. This reduces the calculation (to the same stage) from 7 tables to 4. A second appendix presents calculations and ready-reckoners analogous to those in the body of the paper, but taking account of the temporary (1 Jan. 1967 to 31 Dec. 1968) increase of investment grant from 20 to 25%.

26,725 **Liner Shipping in India's Overseas Trade.** SARANGAN, T. K. *United Nations Publication No. TD B C.4 31, New York*, 1967 [158 pp., 10 ref., 185 tab., 2 graphs, 8 diag.]

This is the first of a series of studies on the shipping problems of selected developing countries, commissioned by the United Nations Conference on Trade and Development. It contains much information on India's overseas trade and its organisation. The main headings are:–

Conference shipping freight rates and the foreign trade of India.
Impact and incidence of liner freight rates on India's exports.
Consultation and negotiation machinery.
Problems related to the improvement and modernisation of ports in India.
Summary and conclusions.

26,726 **Project Cost Estimating.** PETRUSCHELL, R. L. *J. R. Aero. Soc.*, **71** (1967), p. 737 (Nov.) [8 pp., 2 ref., 7 graphs, 6 diag.]

This paper was presented to the Management Studies Group of the Society on 22 June 1967.

The Author, of the RAND Corporation, discusses project cost estimating for long-term military planning (i.e. deciding what weapon and support systems should be developed and introduced), and the relevant concepts and methods of analysis. Conventional cost estimating methods cannot be used to ascertain the resource requirements because the systems to be compared are not known in sufficient detail and because of the uncertainty regarding future conditions. An "analytical" process (often called resource analysis) is adopted: it relies on highly-generalised estimating relationships based on past experience. To illustrate that useful answers to difficult problems can be obtained, even when the uncertainty is great and quantitative formulation is difficult, a particular analysis by the "cost sensitivity" technique is described, with graphs. This relates to variations on the idea of destroying submarine-launched rockets, aimed at the U.S., by counter-missiles launched from patrolling manned aircraft.

The Author considers that the methods discussed will be useful in a much wider field.

Competitive Bidding: Deciding the Best Combination of Non-Price Features. SIMMONDS, K. *Op. Res. Quart.*, **19** (1968), p. 5 (Mar.) [10 pp., 7 ref., 2 tab.]

A procedure is outlined for deciding the " mix " of price and variable non-price features, such as quality, delivery, service, or financing, to be included in a bid. The " level " of each non-price feature is set independently by comparing incremental spending with the alternative price reduction, and basing the choice of level on the value of the feature to the faction in the customer organisation expected to dominate in the choice of the successful bid. Divergence from competition on all non-price features, both fixed and variable, is then taken into account in setting price by calculating a net price equivalent of feature differences as against each competitor. For any given " mark-up ", the probability of success against a competitor over whom there is a net advantage is the same as that for an equivalent lower mark-up using price as the sole basis for allocating the order.

26,727 **Computer Algorithms for Finding Exact Rates of Return [on Investments].** KAPLAN, S. *Journal of Business*, **40** (1967), p. 389 (Oct.) [4 pp., 2 ref.]

SHIPYARDS, DOCKS, AND PRODUCTION METHODS

26,728 **Automatic Location and Cutting of the Ship's Hull Elements.** SIJPANIC, L., and LANDAU, I. *Paper presented at the Symposium on Shipbuilding Automation, Opatija, Yugoslavia*, 22-24 Nov. 1967 [37 pp., 16 ref., 5 graphs, 23 diag., 12 phot.]

The main varieties of flame-cutting machine and associated control systems are reviewed under the headings: Hand control. Magnetic wheel control. Photoelectric control (this system, with 1 : 10 scale lofting, is considered at some length). Numerical control (the discussion of this forms the longest part of the paper).

The aspects of numerical control which receive most attention are:

Information carriers: punched paper tape and magnetic tape; the former is at present more widely used for flame-cutting directors, but the advantages of magnetic tape are explained. They include greater speed and accuracy of data processing, especially where empirical hull curves are involved.

Programming, both manual and by computer: this section includes accounts of the ESSI and of the Eagle systems, which use paper tape and magnetic tape respectively as the final information carrier; also of the " Autokon " system, developed for use with ESSI, which covers all activities connected with the building of a ship (see also Abstract No. 25,440, June 1967).

The paper also contains a section on plasma cutting. This process is known to be successful with alloy steels and non-ferrous materials; research has shown that ordinary steel can be satisfactorily cut with special 50-kW torches using nitrogen as plasma gas and compressed air or oxygen as a jet envelope.

The factors to be considered when choosing a flame-cutting machine, and the measures necessary to ensure its full utilisation in the shipyard, are discussed. Attention should be paid to " nesting " and the utilisation of offcuts; standardisation of plate sizes and of types of edge preparation is also highly desirable.

The Author concludes with some remarks on the rapidly increasing use of computers and numerical control in shipbuilding; automatic draughting machines, and research on the numerical control of bending operations, are mentioned.

26,729 **" Closed Circuit " Grit Blasting.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 829 (14 June) [1 p., 2 phot.]

Cammell Laird (Shiprepairers) have developed a new type of "closed-circuit" equipment for grit-blasting ships' hulls. The process is completely enclosed, and no grit is wasted. The grit is loaded, fed to the blast heads and projected at high velocity through nozzles on to the hull, and returned with scale to the recovery system. Here it is separated from the scale by air washing and then fed back into the system to be re-used for blasting. The method requires considerably less time than conventional methods, and arrangements are included for the treated surface to be prime painted immediately after grit-blasting.

The new plant comprises three mobile grit-blast units: a ship-side cleaning unit and two under-ship units. The ship-side unit is larger than the others, and enables the side of the ship to be treated in almost any position. The blast heads, which are carried on a tower 78 ft high, can traverse either vertically or horizontally down to a height 7 ft 6 in above dock-floor level. No separate staging is required, and the closed-circuit technique ensures the absence of dirt and dust. In one cycle of operation an area of 1 ft 16 ft is treated.

The under-ship units operate similarly, and enable all underside hull areas to be treated up to about 9 ft above the dock floor.

MATERIALS: STRENGTH, TESTING, AND USE

26,730 **Data Sheet No. 179 Steels for Pressure Vessels.** *Engineering Materials and Design*, **9** (1966), p. 969 (June) [1 p., 2 ref., 1 tab.]

A table is given showing the specified chemical compositions and tensile properties (U.T.S., yield strength, elongation) in various B.S. and ASTM specifications and grades for carbon and low-alloy steels. The steel grades concerned are commonly used for cylindrical pressure vessels with welded dished ends. It is pointed out that corrosive fluids may call for a higher-alloy steel, stainless cladding, or a non-ferrous material.

DIESEL AND OTHER I.C. ENGINES

(*See also Abstracts No.* 26,740 *and* 26,769)

26,731 **Effect of Phasing of Compressor and Engine on the Performance of Supercharged Compression-Ignition Engines.** SCHWEITZER, P. H., and WALLACE, F. J. *A.S.M.E., Paper No. 66 WA/DGP 2, presented* 27 Nov.–1 Dec. 1966 [13 pp., 7 ref., 11 tab., 11 graphs, 8 diag.]

The supercharging arrangements of compression-ignition engines often include a reciprocating or rotary positive-displacement compressor, although in many cases aerodynamic compressors are operated in parallel or in series with it. This paper demonstrates, by theoretical analysis and numerical results based on it, that the work input to the compressor depends on the phase relationship between the reciprocation of the compressor and that of the engine. Two-stroke, four-stroke, and free-piston engines are studied, and recommendations made.

The effect of phase is most marked in the case of single-cylinder combinations, particularly the free-piston engine, where the "inward-compressing" engine has unfavourable phasing and the "outward-compressing" engine has favourable phasing. (The latter type is, however, less compact and requires more complex ducting.) The magnitude of the effect depends largely on the volume interposed between the compressor and the engine; in cases of unfavourable phasing, this should be as large as possible. The differences under discussion are most marked in the case of systems employing a pure blowdown turbine (i.e. an exhaust turbine which does not drive a compressor), but the detailed analysis demonstrates the importance of phasing under conditions more frequently encountered in practice.

In an appendix, it is shown how a thermal-efficiency formula for a free-piston engine can be derived without recourse to cycle analysis of the components.

26,732 **Static and Dynamic Tests of Speed Governors for Diesel Engines.** HUTAREW, G., SCHMID, A., and WÜHRER, W. *A.S.M.E., Paper No. 67 DGP 3, presented* 23-27 Apr. 1967 [5 pp., 6 ref., 8 graphs, 7 diag., 1 phot.]

Static and dynamic tests are reported on 14 designs of speed governors for Diesel engines driving mains-supply alternators. The test stand described permitted a constant basic speed to be adjusted between 500 and 1,000 r.p.m. (for static tests), and allowed the superimposition of speed oscillations of variable frequency and amplitude (for dynamic tests). The test results are presented and discussed in the form of coefficients and frequency-response curves.

POWER TRANSMISSION

26,733 **Split Tailshaft Bearings in New Container Ships.** *Motor Ship*, **49** (1968), p. 134 (June) [1½ pp., 1 diag., 1 phot.]

Information on split oil-lubricated tailshaft-bearings developed by the Turnbull Marine Design Co. Ltd (of Sale, Cheshire) was given in the articles summarised in Abstract No. 25,516 (July 1967). The present article gives some further information on Turnbull bearing assemblies of this type (the Mark I), as fitted in three 12,000-ton d.w. container-ships being built by Smith's Dock Co. for Manchester Liners Ltd. (These ships are powered by two geared 7,500-b.h.p. engines and have single controllable-pitch screws.) The bearing seals are a development of Crane Packing Ltd's type 383 seal; the outboard seals have a hot-water supply for de-icing.

See also the following two Abstracts.

26,734 **Further Developments in the Design of Split Tailshaft Bearings.** *Motor Ship*, **49** (1968), p. 181 (July) [1½ pp., 2 diag.]

The Turnbull split tailshaft-bearing assembly (see preceding Abstract) has been further developed, and the Mk. II assembly is now available in ten standard models. These can be delivered as units ("modules") complete with seals and other ancillaries. The bearing proper (of white metal) can be supplied as a "plain" or as a "restricted clearance" bearing. Design data sheets enable the length and diameter of the stern-frame boss to be established as soon as the bearing is selected, so that design of the stern frame can proceed.

The bearing housing of the Mk. II assembly is pressed into the bored boss in the stern frame and welded around the circumference. As with the previous design, the bearing assembly is arranged so that only one man is required for inspecting the bearing and shaft; there are four hydraulic lifting-jacks, as against two in the Mk. I. Inspection, including examination of the lower half of the bearing and closing up afterwards, can be completed in under half a day (e.g. while in port). Other advantages include easy installation and the relaxation of some important design constraints (e.g. a rudder horn can extend further down because the tailshaft can be shipped at an angle; shaft bending moments are reduced because the tailshaft is supported up to the after-flange radius). The tailshaft can have an integral flange-coupling at each end. The seals, which can be inspected and renewed without disturbing the shaft, are Crane Packing Ltd's type 390.

26,735 **Improved Propeller Shaft Seal.** *Design & Components in Engng*, No. 10 (1968), p. 28 (20 May) [1 p., 1 diag., 1 phot.] See also *Shipbuild. Shipp. Rec.*, **111** (1968), p. 372 (15 Mar.) [1½ pp., 2 diag., 1 phot.]

The type 383 seal produced by Crane Packing Ltd. of Slough, is described as a completely new approach to tailshaft sealing. It requires no adjustment and the sealing elements can if necessary be replaced at sea; it can accept longitudinal shaft movements, vibration, and bending.

The seal assembly is basically in two parts, one attached to the hull and the other rotating with the shaft; both parts are split for easy removal. The article briefly describes the construction and functioning of the seal, with the aid of a cut-away drawing; positive pressure at the sealing face is provided by a spring bellows. Should any parts ever need repair or renewal, the shaft can be sealed off from the outside water by means of an inflatable seal incorporated in the assembly.

The seals have been developed to suit all sizes of tailshaft, the largest (at present on test) being one of 56 inches diam. This can accommodate hydrostatic pressures up to 40 lb sq in and shaft speeds up to 105 r.p.m. A photograph shows a type 383 seal being fitted in the *Queen Elizabeth 2*.

See also preceding two Abstracts and Abstract No. 24,074 (Mar. 1966).

26,736 **The Steady-State and Dynamic Characteristics of the Tilting-Pad Journal Bearing in Laminar and Turbulent Flow Regimes.** ORCUTT, F. K. *A.S.M.E., Paper No. 66 Lub 19, presented* 18–20 Oct. 1966 [8 pp., 11 ref., 1 tab., 20 graphs, 2 diag.]

This paper presents design data, with experimental verification, for four-pad tilting-pad journal bearings of equal length and diameter, having a "preload coefficient" of 0 or 0.5 and operating with incompressible lubricants in the laminar and turbulent flow regimes. The turbulent regime and the dynamic properties (stiffness, damping) are of particular interest because such bearings commonly operate at high speeds with low-viscosity lubricants (water, liquid metals). The suitability of tilting-pad bearings for such applications is confirmed. Information on critical speeds, whirling, and stability is included.

26,737 **Standardisation of Gearing for Multi-Engined Propulsion Installations.** MOLLER, F. *Shipbuild. Shipp. Rec.*, **112** (1968), p. 114 (26 July) [4 pp., 5 tab., 8 diag., 1 graph, 1 phot.]

To reduce prod... on costs and hence the price of gearing for multi-engined marine propulsion installations, ASEA are studying the standardisation of gearing. Considerable possibilities exist, and in this article the Author deals with the progress that has been made with twin-engined installations driving one controllable-pitch propeller in the power range 500–3,000 h.p. per engine.

From a survey of existing installations and of current development trends, ASEA have chosen the following characteristics as the basis for a standard series of ships' gears:

Shaft-centres, mm	1,700	2,000	2,300	2,500
Propeller torque, tonne-metres	12	16	21	25
Reduction range . .		2–7·5		
Total output at 200 r.p.m. and reduction 3 to 1, h.p. .	3,400	4,600	6,000	7,000

As regards couplings, the combination of flexible rubber coupling with friction clutch is preferred to the more expensive hydraulic or electric types, and of the possible variations in this type the flange-mounted hydraulically-operated multi-disc clutch is chosen as the standard unit. The propeller thrust bearing is, for reasons of space, lubrication, and installation, connected to the gearcase, but it is arranged so that the propeller thrust is transferred through the bearing housing to its foundation support, the walls of the gearcase not being stressed by the thrust. The standard designs are based also on a completely separate servo system for the c.p. propeller, or a flange-mounted system with a hollow secondary shaft. The designs also embody provision for driving alternators from one of the gear shafts.

After showing sketches of eight possible arrangements of the gears and drives, the Author describes the degree of standardisation already achieved by ASEA for twin-engined installations. Four gearcases of different sizes form the foundation of the series, which comprises, in addition, three sizes of disc-clutch housing, nine of disc clutch, four of input shaft, five of thrust-bearing housing, ten of thrust bearing, three of combined disc clutch and generator gearcase, and three of generator gearing. All components can be used in any of the four gearcases, which are so designed that the input shafts can be above the output shaft. The gearcases all have the same width, and the widths of the gearwheels are standardised in suitable steps so that certain details for one reduction ratio in one gear can, to a certain extent, be used for another reduction in another gear.

Two ASEA gear-lubrication systems are described and illustrated, one for oil feed rates up to 200 l/min and the other for higher rates.

26,738 **Gearing for Multi-Engine Propulsion Installations.** *Mar. Engr.* **91** (1968), p. 93 (Mar.), and p. 123 (Apr.) [8 pp., 1 tab., 3 graphs, 4 diag., 5 phot.]

This is a condensed version of a paper by H. Brauer entitled " The possibilities of gear design to meet the latest requirements of marine propulsion installations". It was presented to a meeting of the Schiffbautechnische Gesellschaft in Berlin

The articles deal only with problems concerning medium-speed Diesel engines with a cylinder output of about 550 b.h.p. at 400–600 r.p.m.; this corresponds to about 8,500 b.h.p. for a single 16-cylinder V-engine or 17,000 s.h.p. with two engines per shaft. Such engines require gears with a reduction ratio between 2·5 to 1 and 6 to 1 for a propeller speed between 250 and 100 r.p.m.

Multi-engine marine installations have been in use for many years, but the earlier drives generally used fluid or electromagnetic slip couplings. Such slip couplings are costly and need considerable space, and a better alternative for the power range considered is provided by a gear unit of straightforward design with mechanical clutch and flexible rubber or laminated-spring coupling. The large North Sea ferry *Tor Anglia*, commissioned in 1966, is an example; she has four 5,580-b.h.p. Diesels driving two propeller shafts through two twin-gear units with 475/250 r.p.m. reduction, giving a speed of 24 knots.

The maximum s.h.p. required at present for merchant ships is about 30,000. This can be provided by four medium-speed Diesels of 7,500 b.h.p. each and a gear unit incorporating four pinions meshing with a bull gear of diameter not exceeding 5 m (16 ft 5 in). Such a bull gear, which can be machined with the necessary accuracy by modern methods, will transmit about 2,260 ton-ft. The pitch-circle speed could be as much as 50 m/s (10,000 ft/min), so that an output of some 40,000 s.h.p. could be safely achieved.

Some possible layouts of single-stage twin-input reduction gears are shown in a diagram; they involve various arrangements of driven, driving, and, in some cases, intermediate idler wheels.

In gear design, calculation of the stresses and loads has to be supplemented by empirical data embodied in the rules of the Classification Societies. The design limits for the teeth are set by the root strength, the surface strength, and the scuffing-load capacity, and a table is given, based on test-bed trials, showing how each of these is affected by geometrical design, material, and type of lubricant. The load-carrying capacity can be substantially increased by various hardening processes.

To achieve the necessary accuracy in machining the gear teeth—some tolerances must not exceed 0·5 micron—the gear-cutting machines are installed in air-conditioned enclosures in which ambient, lubricating-oil, and cooling-oil temperatures are controlled. After machining, the complete gear unit is subjected to a rigorous inspection test, some details of which are given. This test also enables the causes of noise to be established and eliminated. For silent running, not only must the rotating elements be accurately machined, but resonance due to other components, particularly the gear casings, must be avoided. Typical curves are given of excitation frequencies of the internals of a gear unit, and natural frequencies of the casing are plotted against propeller speed.

To absorb the propeller thrust, the general practice is to incorporate the thrust bearing in the gear unit and to couple the propeller shaft to the gear output shaft. It would be better, but more expensive, to arrange the thrust bearing separately, mainly to isolate the bull gear from the pulsating axial forces of the propeller shaft

Rigidity of the gear seatings is important, particularly adequate torsional rigidity. A closed box girder without access holes best meets these requirements.

The second article is concerned with couplings and clutches. With medium-speed Diesels, couplings should have torsional flexibility to damp out the torque variations which, although they may be limited to 10 to 15% of the mean at full speed, may be much greater at reduced speed. The requirements can be met by rubber couplings or laminated-spring couplings. The nominal torque of a typical 8,500-b.h.p. engine of this type is about 15 metre-tons, and the largest rubber coupling now in production can transmit 55 metre-tons.

Friction clutches can be either wet or dry. The problem with both is mainly one of heat dissipation, and in this respect the wet clutch has the advantage. It is also less subject to wear because of the lower coefficient of friction between its working surfaces. For this reason the contact load on the rubbing surfaces must be greater than for the dry clutch. In multi-plate form, wet clutches can now accept specific pressures of 30 kg/sq cm (425 lb/sq in), and relative speeds between the rubbing surfaces of 40 m/s (7,875 ft/min).

The working conditions imposed on the clutch during reversing or rapid manœuvres of the ship are discussed. Three cases are considered, namely, reversing clutches for use with non-reversing engines and fixed-pitch propellers, clutches for twin installations with reversible engines and fixed-pitch propellers, and disconnecting clutches for engines with controllable-pitch propellers. The transmission capacity in all cases is limited by the surface temperatures of the rubbing faces and the heat dissipation. As an example, the calculated temperature rise is 25°C (45°F) during the reversing of a ship with four non-reversible engines of 2,200 b.h.p. each at 900 r.p.m., driving two reverse reduction gear units.

It is shown that some considerable time elapses during a properly executed reversing manœuvre, and that this can possibly be reduced by the use of a shaft brake.

LAYOUT AND INSTALLATION

26,739 Forced Hull-Vibration and Rational Alignment of Tailshafts (in French). WOJCIK, Z. C. *Bull. Tech. Bur. Veritas*, **50** (1968), p. 21 (Feb.), p. 45 (Mar.), and p. 77 (Apr.) [44 pp., 44 ref., 1 tab., 20 graphs, 30 diag., 20 phot.]

Increases in the size and power of ships since the second World War have resulted in hulls which are more flexible and propeller shafting which is more rigid, a combination which has led to stern-tube bearing and tailshaft troubles and to hull-vibration problems. The Author discusses these interrelated difficulties with particular reference to their relationship with tailshaft alignment. He suggests that they can be overcome by a "rational" alignment of the tailshaft, i.e. an alignment which ensures that the tailshaft cannot lose contact with the lower parts of its bearings, either under the influence of static loads (due to the weight distribution of the line shafting, or to the hull deformations encountered in service) or under the combined influence of static and dynamic loads (due to excitation by the propeller, by the flexural vibration of the shafting or tailshaft, or by the free and/or forced vibration of the hull-girder). (See also Abstracts No. 24,809, Nov. 1966, and 25,353, May 1967.)

Rational alignment involves abandoning conventional straight-line alignment and parallel concentric couplings. The Author describes a method of calculation (with the aid of a computer) for obtaining the necessary data for rational alignment, together with practical methods of carrying out such alignment.

The information, which is based on Bureau Veritas experience and research, is presented under the following headings:

1. Introduction.
2. Behaviour, in Practice, of the Tailshaft Bearings Stuffing-box Assembly.
   - (A) Stern-tube with lignum vitae bearings.
   - (B) Stern-tube with white-metal bearings.
3. Experimental Research and Theoretical Considerations in the Investigation of the Phenomena.
   - (A) Experimental research on tailshaft behaviour.
   - (B) Theoretical considerations relating to tailshaft behaviour.
     - (i) Distribution of static reactions due to the weight of the shafting assembly.
     - (ii) Influence of cargo-loading conditions, and of sea states, on values of static reactions.
     - (iii) Hydrodynamic effects of propeller, and their influence on forced hull-vibrations and on tailshaft behaviour.
     - (iv) Variations in bearing-reaction values due to flexural vibration of the shafting.
4. Calculations for Rational Alignment.
   - (A) Review of the problem, and the concept of straight-line alignment with parallel concentric coupling-flanges.
   - (B) The rational-alignment concept.
   - (C) Carrying out the necessary calculations.
     - (i) Preparing the data for the calculations.
       - (*a*) Description of the " beam ".
       - (*b*) Loading of the " beam ".
   - (D) Example of calculations for rational tailshaft-alignment.
5. Carrying Out Rational Tailshaft-Alignment in Practice.
   - (A) Optical method of alignment.
   - (B) The " uncoupled-flanges condition " method of alignment.
6. Conclusions and Suggestions.

See also Abstract No. 23,886 (Jan. 1966).

LUBRICANTS AND LUBRICATION

26,740 **Cylinder Lubrication of Large Marine Diesel Engines.** Hosh, R. M., and Schrakamp, J. W. A. *Motor Ship*, **49** (1968), p. 149 (June) [3½ pp., 7 ref., 2 tab., 3 diag., 1 graph]

The importance of efficient cylinder lubrication has been enhanced by the two main modern developments in large marine Diesel engines the use of heavy fuel oils and the very high power outputs per cylinder now being sought.

The problems resulting from burning heavy fuels are mainly associated with the high sulphur content of these fuels. Early work showed that the alkalinity of additive-type oils had to be greatly increased to combat the enormous increase in corrosive wear and fouling. This led to the introduction in 1956 of a highly alkaline emulsion-type cylinder lubricant and in 1959 to the now more common alkaline single-phase oils.

In addition to its normal function of preventing metal-to-metal contact between piston and cylinder, the lubricant has to cover the entire swept surface with sufficient alkalinity to protect it against corrosive attack by the acidic products of combustion of high-sulphur fuels. The reciprocating piston distributes the lubricant rapidly in narrow axial bands on either side of the supply points. The circumferential movement is much slower and, unless suitable precautions are taken, large areas of cylinder surface between the oil-supply points may not receive sufficient alkaline additive. To prevent additive starvation in these areas attention must be paid to the level of alkalinity in the cylinder oil, the oil-supply arrangements, and the engine operating temperatures. Each of these items is discussed by the Authors, as well as certain other engine factors that affect lubrication.

Alkalinity level. Laboratory tests, confirmed by 16 years of experience at sea, have shown that, for marine engines, an alkalinity level (in terms of Total Base Number, TBN) of at least 60 mg KOH per gram and preferably somewhat higher, is required for the burning of residual fuels containing up to 4% by weight of sulphur. An increase in the rate of feed of oils of lower alkalinity can lead to other problems such as exhaust-port blockage, scavenge-belt fires, and excessive deposits on the pistons.

Oil-supply arrangements. The oil-supply points should be evenly spaced around the circumference of the liner and the distance between them should not exceed 38–40 cm (15–16 in). The latest engine types now under construction, with cylinder diameters of 850–1,050 mm (31·7–41·3 in), therefore need no more than 8–10 supply points. If this number is increased to more than about 12, the quantity of oil required at each point becomes so small that metering it becomes difficult.

Oil-spreading grooves, when properly designed, can improve conditions in cylinders with unevenly-spaced oil-supply points. Details of an oil-groove system designed by Shell are shown in a diagram.

The axial positions of the supply points are also of importance. There is evidence that, at least for highly supercharged engines, the best location is in the lower part of the stroke. Further, there should be effective non-return valves in the oil-supply lines as close to the liner as possible, to prevent hot combustion gases from entering the lines.

The correct feed rate of the cylinder oil is not necessarily that corresponding to the least cylinder wear; it is the rate that gives the most economical engine-room. Overhaul expenses such as the cleaning of pistons and ports must be taken into account as well as the cost of oil and of piston-ring and liner replacements.

As regards wear, feed rates must be increased as the thermal load on the engine increases; and large engines with thick liners need more oil than smaller engines with thinner liners because of the higher surface temperatures. A rate of 0·4–0·5 g/b.h.p.-hr is generally sufficient for highly-rated modern designs of uniflow-scavenged engines. This limits liner wear to about 0·04–0·05 mm (1·6–2·0 mil) per 1,000 hours. For loop-scavenged engines of similar dimensions and output the rate should be 0·6–0·7 g/b.h.p.-hr. Lower feed rates may sometimes be satisfactory, but the margin of safety will be substantially reduced if these rates are much reduced. An indication of whether the lubrication is satisfactory can be obtained from chemical analysis of the cylinder-oil drainings. These should show a TBN value of not less than 10 mg KOH/g.

Frequency and timing of oil delivery are also important. If delivery is too infrequent the oil film may be thinned too much by evaporation, and if inappropriately timed much of the oil may be swept out of the ports or even burnt.

Engine operating temperatures. In most modern engines, the outlet temperature of the cooling water is 65–70° C (149–158° F), which, together with a TBN number of the lubricating oil of over 60 mg KOH/g, ensures a satisfactorily low wear rate. With the increasing thermal loads and power output now possible with large, high-pressure supercharged engines, there is a tendency to run at a lower cooling-water temperature, but this should not be lower than 60° C (140° F) at outlet.

Other engine factors affecting lubrication. A number of cases of very high wear have been caused by inadequate side clearance of the piston rings. It is important to ensure that the clearance is at its correct design value, especially during overhaul in service. The effect of low ring side clearance will be accentuated by any piston distortion, and this is another factor that cannot be ignored in the design of large-bore, supercharged, high-output engines.

At sea, the engine can be temporarily overloaded, e.g. in bad weather by an overpitched propeller, and cyclic variations have been observed in cylinder-liner temperature of 20–40° C (45–72° F) above the steady value recorded on test-bed trials. The liner temperature may thus become too high for mineral oils, and the safety margin will therefore be lowered.

Good maintenance of the fuel injector is of great importance. The cleanliness and correct viscosity of the oil before injection must be closely controlled.

AUXILIARY EQUIPMENT AND MACHINERY

(*See also Abstracts No.* 26,772 *to* 26,775)

26,741 **Piston-Engine Heat Recovery Systems—Design and Selection.** Cusy, P. E. *A.S.M.E., Paper No. 67–DGP–1, presented* 23–27 Apr. 1967 [5 pp., 1 tab., 3 graphs, 3 diag.]

This paper is concerned with heat recovery from exhaust gas and jacket-cooling water. The first section explains the considerations involved in designing heat-recovery silencers, steam separators, and related equipment. The second section considers the principles governing selection of equipment for particular installations; special attention is paid to control and safety arrangements. A nomogram is given for pipe sizing in natural-circulation systems.

26,742 **An All-Fresh Water Cooling System.** *Mar. Engr.* **91** (1968), p. 130 (Apr.) [1 p., 1 diag.]

To reduce the heavy maintenance costs of large salt-water cooling systems, Eriksbergs Mek. Verkstads have designed a three-circuit cooling system for the main and auxiliary machinery of the 107,000-ton d.w. tanker *Kungaland*. The main machinery consists of a ten-cylinder 840-mm bore Eriksberg-B. & W. engine rated at 25,300 h.p., and fitted with four Brown Boveri turbochargers and Serck charge coolers. There are three 466-kW Diesel-driven generators and a 500-kW steam-turbine driven alternator.

The cooling system is divided into three circuits: salt water, low-temperature fresh water, and high-temperature fresh water. Salt water is used only to cool all the fresh water in a very large central heat exchanger, and in a vacuum condenser serving the turbo-alternator and cargo-oil pump turbines. For circulating the salt water there are a pair of 1,000 tons hr pumps, either of which is able to meet all cooling needs except when the sea temperature exceeds 30 C (86 F), and the cargo or ballast pumps are in use. The two fresh-water circuits use the same water and have a common expansion tank, but they are maintained at different temperatures. The low-temperature circuit, maintained at 35 C (95 F), cools all the machinery in the engine room except the main and auxiliary engine jackets, which are served by the high-temperature circuit maintained at an inlet temperature of 38 C (100 F). Both fresh-water circuits contain corrosion inhibitors.

A diagram shows the arrangement of the three cooling circuits.

26,743 **Exhaust Boilers on Ships Regulation and Automation Problems and Solutions.** DURIC, V. *Paper presented at the Symposium on Shipbuilding Automation, Opatija, Yugoslavia*, 22 24 Nov. 1967 [30 pp., 1 graph, 5 diag.]

This paper is based on the Author's experience at the " Duro Daković " factory, which has built many exhaust-gas boilers and oil-fired boilers for motor ships (especially Soviet tankers). Emphasis is laid on the differing characteristics of exhaust-gas boilers with and without gas by-pass arrangements. A by-pass with diverter valves (which may be automatically controlled, e.g. by a steam-pressure signal) gives greater operational flexibility and permits more satisfactory control; in particular, steam conditions can be kept constant at all loads. Boilers without a by-pass are, however, becoming popular on account of their simplicity and compactness; the relatively low exhaust temperatures (e.g. 715 F) of low-speed engines permit " dry " operation when no steam is required. The design and operational problems of these two types of boiler installation are discussed at length, and suitable control arrangements are described. Special attention is paid to the feed system in boilers without by-pass; operation of such a boiler in parallel with an oil-fired boiler is also considered.

A set of heat-accumulation curves for a by-pass installation in the tanker *Split* (20,800 tons d.w.) is included.

26,744 **A British-Designed Filtration Unit for Heavy Fuel.** *Motor Ship*, **49** (1968), p. 189 (July) [1½ pp., 2 diag.]

A description is given of a new automatic heavy-fuel filter module, the Series 1000, produced by Vokes Ltd. Its type designation denotes its capacity of 1,000 gal hr; other sizes can be made available. Contracts for this filter module are being discussed with several U.K. and other shipowners.

Basically, the Series 1000 filter module consists of a duplicated pump set, a heat exchanger, a primary filter, a duplex secondary filter, and a control system; the assembly is mounted as a unit on a base serving as a drip tray. For marine installations, manually-operated valves will be provided to enable the module to be by-passed or to allow recirculation and warming-up. The pump set selected for duty would pass the incoming oil to the heat exchanger for heating to the required temperature; the filter bodies are steam-jacketed, but alternative means of heating them can be provided. From the heat exchanger, the oil passes to the primary-filter section for water removal and initial filtration. This section has three filter-elements working in parallel, and provides for dumping of collected water, element cleaning (one element at a time), and dumping of collected dirt. The cleaning operations are pneumatically powered, and are automatically controlled by an electric programming unit. On leaving the primary filter, the oil goes to the secondary filter, which consists of two Microfelt 200 elements in separate casings; when the on-line element requires changing, a pneumatic actuator automatically cuts it out and brings the other element on line without interruption. The secondary stage filters down to 5 microns.

The article gives further information on the design and operation of this filter module, and mentions that the increasing acceptance of unattended engine rooms has heightened interest in fuel purification by filtration instead of by centrifuging.

26,745 The Application of Screw Pumps for Tanker Cargo Handling. BLEIJENBERG, G. W. *Motor Ship*, **49** (1968), p. 187 (July) [2 pp., 3 diag.]

The variety of chemical and other liquid cargoes now being carried in tankers, and developments in the design of screw pumps, have resulted in this type of pump being increasingly used for cargo handling. Screw pumps with a capacity of 1,500 tons/hr can now be supplied.

Within the viscosity range 300 to 1,500 sec. Redwood No. 1, the capacity of these pumps remains substantially constant. Products with a viscosity above 1,500 sec. can be discharged at the same rate if the suction line is large enough. The efficiency of the screw pump, when handling high-viscosity liquids, is considerably higher than that of other types.

As a result of improved bearings and screw profiles, screw pumps can provide discharge pressures of up to 200 lb/sq in; special designs can provide up to 350 lb/sq in. The improved profiles give very good efficiency irrespective of viscosity. Improved mechanical seals ensure freedom from leakage at the shaft, under severe conditions of temperature, vapour pressure, and viscosity. Casings can be of highly corrosion-resistant materials, such as AISI 316 (EN 58 J) and Hastelloy.

The Author, of Houttuin Pompen NV, Utrecht, summarises the advantages of the screw pump as: (*a*) The suction characteristics are such that, even when liquid and vapour are handled together, the pump does not lose suction; this is of great importance when products of high vapour pressure are carried and the tanks must be fully drained, and frequently enables an independent stripping pump to be dispensed with. (*b*) As the pump capacity is not sensitive to viscosity changes, discharging times for different cargoes can be maintained at the same level. The Author also discusses the various factors to be considered when designing the pumping system, including the properties of the liquid, pump position, drive arrangements, and stripping.

A brief description is given of a " pump combination " for products whose very high vapour pressure necessitates the pump being placed within the tank to ensure that the permissible low suction lift is not exceeded. In this arrangement, a vertical screw-pump is mounted on deck and fed by a small axial-flow centrifugal booster-pump fitted at the foot of the suction pipe. The booster pump is shaft-driven off the screw pump, which is itself driven by a motor (preferably hydraulic rather than electric). This pump combination is suitable for products with a viscosity of up to 3,000 sec. Redwood I and having a vapour pressure of 11·5 lb sq in. Apart from eliminating the need for a pump room, the arrangement minimises toxicity hazards and reduces the number of lines and valves; as each tank has its own pump combination, there is little chance of contamination of one product by another.

AUTOMATION, INSTRUMENTS, AND CONTROL DEVICES

(See also Abstracts No. 26,732 and 26,743)

26,746 **Remote Control System for Two Schelde-Sulzer 6RD90 Engines.** VERKLEY, C. J. *Holland Shipbuild.*, **16** (1967), p. 48 (July) [3 pp., 2 diag., 4 phot.]

Two 45,000-ton d.w. bulk carriers, the *London Bridge* and the *Forth Bridge*, both owned by the Bowring Steamship Co., London, are each fitted with a Schelde-Sulzer 6RD90 engine of 13,800 b.h.p. at 119 r.p.m. The engine can be controlled either from a control room in the engine room by a mechanical system using Flexball cables, or from the bridge by a Sulzer-Westinghouse pneumatic system incorporating standard pneumatic elements.

The movement sequence of the different servomotors is programmed by pneumatic relays and timing elements which are grouped in a cabinet in the control room. All commands from the bridge are transmitted by three ⅜-in copper pipes to the cabinet, from where other pipes lead to the manœuvring stand with its servomotors and thence to the main engine.

The functions of the main components of the pneumatic and mechanical remote-control systems are described with the help of a diagram. The sequence when starting the engine for the " running ahead " condition by means of bridge control is also described in detail. A " jump over " device (incorporating a micro-switch operating a solenoid valve) ensures rapid acceleration or deceleration through the barred speed range of 60 to 72 r.p.m.

The operation of the system during a crash stop when under bridge control is described in detail. Briefly, it involves placing the telegraph lever in the astern position, which causes the fuel supply to be cut off and the engine to be braked by the starting air. A direction-of-rotation safeguard control valve is incorporated in the system. To change from bridge to engine-room control, the engine-room telegraph handle is moved from the special "bridge control" position to any other desired position; this causes the pneumatic system to be vented and put out of action.

There are photographs of the bridge and engine-room control stations and also of the contents of the pneumatic cabinet.

26,747 **Liquid Level Indicators. Part 1 Visual and Mechanical.** GELLETT, C. **Part 2 Level Indication by Electronic and Electrical Means.** DAVIES, M. A. S., and LAZENBY, B. D. *Engineering Materials and Design*, **11** (1968), p. 390 (Mar.), and p. 559 (Apr.) [14 pp., 2 tab., 5 diag., 13 phot.]

The types of level indicator discussed in Part 1 are: Visual (windows, sight glasses, etc.). Float arm driving dial pointer. Dipstick and related devices (e.g. dip tape, drip-stick, and right-angled swivelling tube for pressurised tanks). Gauge head (guided float linked by wire or tape to automatic reel at top of tank). Magnetic (the incorporation of a magnetic link in float-type devices enables the indicating mechanism to be isolated from the liquid; accurate " magnetic dipsticks " and " magnetic gauge heads " using this principle have been developed). Hydrostatic (head measurement by pressure gauge; various means of remote reading are available). Part 1 concludes with a discussion of the factors to be considered in selecting the right instrument for a given application. Most manufacturers will supply their instruments calibrated to customers' requirements (depth, volume, etc.). It is pointed out that these visual and mechanical instruments require very little maintenance and, in most cases, are independent of external power supplies.

The indicators described in Part 2 rely mainly on electrical or electronic equipment; they therefore have an inherent remote-reading capability and lend themselves to use in automatic control systems. Most of them can be used with powders, granular solids, and slurries as well as liquids. The types described are as follows: Capacitance. Ultrasonic. Radioactive-isotope. Displacement (in this system a float is supported partly by its own buoyancy and partly by a torsion spring or torque tube, the latter permitting a welded seal for high-pressure applications. Although the system is essentially mechanical, it is often designed with electrical or electronic signal transmission). Balance or " precise tank gauge " (this covers a variety of systems using sensing devices of very different types, e.g. float, capacitance, transmitting aerial; the device is supported by a tape or wire which is automatically wound in or paid out to maintain a condition of balance. High accuracies can be achieved). Load cells (these measure the total weight of the tank and its contents, from which information the level can be estimated). Pressure transducers. Multiple Discrete (this term covers systems whose common feature is a vertical sequence of above below sensors; many different types of sensor can be used for this purpose). Each type of indicator is described in sufficient detail to enable a plant engineer to decide on the most suitable system in relation to the hygienic, corrosive, and abrasive properties of the liquid or solid concerned, the type of container, the temperature and pressure conditions of the contents, the " intrinsic safety " or flameproof condition of the environment, and so on.

The article concludes with a table listing 84 U.K. manufacturers and or suppliers of liquid-level indicators, and showing the types available from each of them. There is also a supplementary list of 13 further U.K. manufacturers, etc.

26,748 **Survey of Pressure Transducers.** *Engineering Materials and Design*, **11** (1968), p. 221 (Feb.), and p. 397 (Mar.) [9 pp., 9 ref., 2 tab., 1 graph, 6 diag., 10 phot.]

This survey gives detailed accounts of some widely-used types of pressure transducers and associated equipment, under the headings: Sources of information on pressure transducers. Definitions (of terms relating to performance characteristics some terms in general use are listed and explained). Potentiometric transducers. Examples of potentiometric transducers. Strain-gauge transducers. The differential-transformer type. Frequency modulation. Force balance transducers. Digital transducers.

There is a table giving the names and addresses of 31 U.K. pressure-transducer manufacturers and agents, together with the principal features of their products; also a supplementary list of 37 other manufacturers and suppliers.

26,749 ***Correction of Errors in Indicator Diagrams: Passage Effects.*** BRADLEY, P. G., and WOOLLATT, D. *Engr.* **225** (1968), p. 511 (29 Mar.) [4 pp., 4 ref., 1 tab., 26 graphs, 3 diag.]

The Authors describe a simple approximate method whereby unsteady pressure diagrams taken with a long indicator passage may be corrected. The relevant theory is set out at length. Experiments using the pulse generator referred to in Abstract No. 21,824 (Aug. 1964) are described and the results presented. They show that the correction method gives a considerable improvement and might, in many cases, make otherwise useless results acceptable. In some situations it may now be preferable to use a passage rather than a flush-mounted transducer.

DECK MACHINERY AND CARGO HANDLING

26,750 ***Transfer of Oil Cargo at Sea.*** *Shipbuild. International*, **11** (1968), p. 28 (July) [2½ pp., 8 phot.] **Shell's " Lightening at Sea " Scheme.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 819 (14 June) [3 pp., 5 phot.]

These two articles deal with the scheme developed by Shell International Marine Ltd for transferring to a smaller tanker at sea enough of the oil cargo of a V.L.C.C. (very large crude carrier) to enable the latter to enter a port unable to accommodate her full draught.

The original design of the Shell V.L.C.C.s was for tankers of about 165,000 tons d.w. with fully-laden draught limited to 54 ft, i.e. the depth available at most European ports. It was known, however, that plans were afoot to deepen several of these ports to take ships of over 200,000 tons d.w. Europort, Le Havre, and Gothenburg by early 1970 and Foz a little earlier; and investigation showed that it would be economically beneficial to increase the size of these new tankers to 200,000 tons d.w., with a draught of 62 ft, even if they had to operate for the first few years of their life only partly laden until the ports were ready. At a draught of 54 ft a V.L.C.C. of 200,000 tons d.w. carries about 170,000 tons of crude oil.

Twenty-two of these 200,000-ton d.w. V.L.C.C.s (known as the M-class; see also Abstract No. 26,537, July 1968) have been ordered and seven more are to be chartered. Some are already in Shell service. All will operate mainly between the Middle East and Western Europe.

Realising that there would be considerable economic advantage if it were possible, before the ports were deepened, for these ships to sail fully-laden and to transfer part of their cargo at sea to another tanker outside their destination ports, Shell International Marine carried out on paper a six-months simulation study of the movements of V.L.C.C.s operating such a "lightening" schedule. The data taken into account included weather information and forecasts, the contract delivery dates of new ships, details of voyages to fit in with crude-oil movements from the Middle East to Europe, oil supply and demand estimates, and other relevant factors. The results showed that the scheme was well worth while: the net saving from 20 ships making five voyages each per year would be about £12·5 million.

To put the scheme to practical test, the 70,000-ton tanker *Drupa* (see Abstract No. 25,226, Apr. 1967) was modified in the summer of 1967 to serve as a lightening vessel, and in March 1968 the first sea trial took place with the 207,000-ton d.w. *Macoma* of Shell Tankers N.V., Rotterdam. Sea water was used instead of oil in this operation, which the two articles describe, giving details of the fenders, hoses, mooring equipment, and pumping procedure. The rate of transfer was about 6,000 tons/hr, including approach, level-up, and break-away manœuvres, which represents a transfer of 70,000 tons in a total of about 16 hours.

The first transfer of crude oil took place on 30 May, when 65,650 tons of crude oil from the Middle East were transferred from the *Macoma* to the *Drupa* in a position eight miles off Berry Head. After the two tankers uncoupled on 31 May, the *Macoma* sailed for Europort and the *Drupa* for Thameshaven.

The six-month simulation exercise showed that two lightening vessels would be required for full operation, and a second tanker is accordingly being modified.

26,751 **Containers and Safety.** BUXTON, G. H. E. *A.S.M.E., Paper No. 67 TRAN 37, presented* 28 30 Aug. 1967 [*7 pp., 13 ref., 7 phot.*]

Figures are quoted to illustrate the rapidly-growing use of containers in the shipping industry. It is noted that the accident frequency at certain U.S. container terminals appears to be less than one-quarter of that for break-bulk cargo handling. Progress in the standardisation of container designs is reviewed. Attention is drawn to the possibility of dissimilar-metal corrosion, as most metal containers incorporate both aluminium and steel.

The various classes of ship, ranging from the "pure" container ship to the conventional general-cargo vessel, in which containers are at present loaded, are reviewed, and the handling and stowage arrangements normally adopted in each case are noted. Most container failures have occurred while the container was suspended; in some cases the roof was torn away and in others the container buckled and sometimes split. Possible causes are weakness of corner posts and/or corner fittings; incorrect application of loading slings; bad distribution of cargo weight inside the container (this also affects the stability and safety of ships measures must be taken to ensure correct stowing and securing of loads within containers). The possibility of containers being damaged while stowed in cells (e.g. by very severe ship motions), and the more likely case of damage during movement into or out of a cell, must be considered and equipment devised for dealing with such damaged containers.

Planned maintenance schedules and periodic inspections of containers should be arranged; individual checks should be made as they enter the terminal. Some miscellaneous safety problems are briefly considered. These include foothold on the tops of containers in rain or ice conditions; the extent to which nature of contents should be indicated on the outside; safety in securing containers on deck by manual methods; the large area presented to the wind by deck-stowed containers; and the need for devices to avoid guiding suspended containers into cells by hand (this has sometimes been necessary owing to an unfavourable combination of ship trim and container-handling gear).

VIBRATION AND SOUND-PROOFING

(*See also Abstracts No.* 26,713 *and* 26,739)

26,752 **Balancing of High-Speed Machinery.** HAN, C.-Y. *J. Engng Industry*, **89** (1967), p. 111 (Feb.) [7 pp., 6 ref., 3 tab., 2 graphs, 7 diag.; and Discussion: 1 p., 4 ref.]

This is A.S.M.E. Paper No. 66-Mech-29, presented 10-12 Oct. 1966. An analytical method for optimum balancing of the shaking forces and the shaking moments of force of any plane mechanism driven by a constant-speed shaft is developed. The optimum values of phase angle and mass moment of a balance weight on the driving shaft are ascertained, the criterion being that the variation of load on the machine supports due to shaking forces and shaking moments of force shall be a minimum through a complete revolution of the driving shaft. With a digital computer, this method is applicable to complicated mechanical systems; an illustrative example relates to the balancing (for minimum vibration) of a punch/reader unit in the idling condition.

26,753 **Calculations and Experiments on the Unbalance Response of a Flexible Rotor.** LUND, J. W., and ORCUTT, F. K. *A.S.M.E., Paper No. 67-Vibr-27, presented* 29-31 Mar. 1967 [12 pp., 11 ref., 3 tab., 16 graphs, 6 diag.]

The results of a combined analytical and experimental investigation of the unbalance vibrations of a rotor are presented. The analysis applies to a general rotor bearing system in which the dynamic bearing forces are represented by four spring coefficients and four damping coefficients. The rotor can be represented as either a lumped or a distributed parameter system, and gyroscopic moments are included. In general, the unbalance whirl motion of the rotor will be elliptical. The analysis has been programmed for a digital computer, to obtain results for comparison with the experimental data.

The test rotor is a uniform flexible shaft with heavy wheels mounted at the ends and in the middle. It is supported in two silicone-fluid lubricated tilting-pad journal bearings. The rotor amplitude caused by an induced unbalance has been measured over a speed range of 3,000 to 24,000 r.p.m. for three different rotor configurations, obtained by removing one or both end wheels. This speed range extends to or through the third critical speed for each of the rotor configurations. The results are compared with the theoretical values and, in general, the agreement is found to be good. It is concluded that rotor response calculations can be of great practical value in the design of high-speed rotor bearing systems.

26,754 **On the Critical Speeds of a Continuous Shaft Disc System.** ESHLEMAN, R. L., and EUBANKS, R. A. *A.S.M.E., Paper No. 67 Vibr 9, presented* 29 31 Mar. 1967 [8 pp., 4 ref., 1 tab., 9 graphs, 3 diag., 1 phot.]

The effect of gyroscopic moments on the critical speeds of a shaft/disc system mounted in short end bearings is analysed. Representation of the shaft as having continuously distributed mass and elasticity allows accurate determination of higher critical speeds. Frequency equations are obtained for the critical speeds associated with both backward and forward whirling modes. The first four critical speeds for each whirling direction are shown graphically for a range of shaft and disc sizes and for various disc locations on the shaft. Experimental verification is given for the first and second critical speeds, and comparisons are also made with results obtained by lumped-parameter methods.

26,755 **Coupled Vibration of Geared Systems.** MAHALINGAM, S. *The Aeronautical Journal (formerly J. R. Aero. Soc.),* **72** (1968), p. 522 (June) [5 pp., 10 ref., 1 graph, 5 diag.]

The standard methods for analysing the torsional vibration of geared systems assume that the rotating system is rigidly supported. However, in many practical applications the gearbox (or the crankcase or frame to which it is attached) is flexibly mounted, and coupled vibrations of the rotating system and the supporting system can arise. The Author presents an analysis of this situation, using the concepts of "free receptance" and "support receptance" (introduced by S. H. Crandall in 1958), and receptance formulae of Biot-Duncan type (see also Abstract No. 1,345, Nov. 1947). It is shown (with examples) how the influence of mounting flexibility on natural frequencies may be calculated

An appendix shows how the Biot-Duncan formula can be extended to the cross receptances of a torsional system.

26,756 **Torsional Vibration of a Geared System.** HO CHONG LEE, and STEVENSON, C. H. *A.S.M.E., Paper No. 67 Vibr 63, presented* 29 31 Mar. 1967 [5 pp., 2 tab., 1 diag.]

A method of computing torsional natural frequencies and steady-state responses of a geared system is presented and discussed. The system consists of a number of shafts, each with any number of masses, some of which are gears to interconnect the shafts. A shaft may have any number of gears which branch out to other shafts. For response calculations, harmonic torque may be applied to any of the masses. To allow the most general combination of branching systems, the method uses a computer program which forms a matrix system from the influence coefficients of each shaft due to unit torque applied at the gears. The unknown meshing torques form the column matrix, and the matrix equation is solved to find natural frequencies and mode shapes. The method easily handles damped vibrations by the use of complex values for the inertia and elasticity. A numerical example is given, with specimen print-outs.

26,757 **Recent Blade Vibration Techniques.** ARMSTRONG, E. K. *A.S.M.E., Paper No. 66 WA GT 14, presented* 27 Nov. 1 Dec. 1966 [8 pp., 7 ref., 2 tab., 8 graphs, 1 diag., 5 phot.]

This paper is based on the experience and techniques of Bristol Siddeley

Engines Ltd in the design and development of gas-turbine engines.

The methods which have been used recently to predict the amplitudes of vibration of compressor blades are explained. Examples of resonances with excitation due to maldistributions in the intake flow, and to downstream obstructions, are given. Designs of tied blades and snubber blades which have operated satisfactorily are illustrated and briefly described. Two techniques of obtaining blade vibration data from the H.P. shaft of a two-shaft jet engine are described; these are the "FM grid" technique of Eccles and Seymour, which uses the signal induced by a blade-tip magnet in a zig-zag stator conductor, and the radio-frequency strain-gauge telemetry system referred to in Abstract No. 23,990 (Feb. 1966). By comparing the measured amplitudes with the fatigue properties of blading, a parameter has been established which is used in assessing the seriousness of a vibration in relation to required service life of the blading.

Further work should concentrate on prediction of flutter conditions; fatigue strength of root fixings; the importance of mechanical damping as a means of controlling blade vibration; and design-stage prediction of blading fatigue strength.

26,758 **Whirl in Reciprocating-Engine Flywheel Crankshaft Systems.** Lowell, C. M. *A.S.M.E., Paper No. 67 Vibr-59, presented* 29–31 Mar. 1967 [12 pp., 5 ref., 2 tab., 15 graphs, 26 diag.]

The coupling of torsional vibration and whirl in reciprocating-engine flywheel crankshaft systems was discussed in 1959 by the present Author; a series of laboratory tests established the existence of this coupling. Since then, large amounts of field data have been obtained and analysed, with the result that the phenomenon is much more completely understood. This paper reviews the theory, describes and discusses the field studies, and presents a rational design procedure which should prevent catastrophic whirl resonance and consequent crankshaft failure.

26,759 **Composite Damping of Vibrating Sandwich Beams.** DiTaranto, R. A., and Blasingame, W. *A.S.M.E., Paper No. 67 Vibr 6, presented* 29–31 Mar. 1967 [6 pp., 4 ref., 1 tab., 6 graphs, 1 diag.]

Laminated beams composed of alternate layers of elastic and visco-elastic material have been considered for structural members which can dissipate vibratory energy while maintaining a degree of structural integrity. The vibratory characteristics of such beams have already been investigated by the Authors and by other workers. DiTaranto has previously shown that, for a freely vibrating three-layer (elastic-viscoelastic-elastic) beam, the curve of composite loss factor versus natural frequency is independent of the end conditions and the mode shapes. The present paper investigates the solution of the differential equation of motion, in order to obtain generalised results for the composite loss factor and natural frequencies of such a sandwich beam. These results (which are presented graphically) can be useful in designing structures utilising sandwich-laminated materials.

26,760 **Damping in Sandwich Beams with Shear-Flexible Cores.** Bert, C. W., Wilkins, D. J., and Crisman, W. C. *A.S.M.E., Paper No. 67 Vibr 11, presented* 29–31 Mar. 1967 [9 pp., 32 ref., 6 tab., 8 graphs, 2 phot.]

After a literature survey, this paper reports the Authors' theoretical and experimental studies of the effect of core shear flexibility on the lowest natural frequency, node locations, and damping in sandwich beams with cores of high shear flexibility (e.g. honeycomb-type cores). A new method of analysis is presented for predicting the logarithmic decrement for damping in sandwich beams undergoing free vibration, when the beam geometry and constituent-material properties are known. Natural frequency, modal shape, and logarithmic decrement all depend on the dynamic shear coefficient. Two new simplified derivations for this coefficient are presented.

Flexural-vibration experiments were conducted on free-free sandwich beam strips at frequencies from 300 to 700 c.p.s. Facings were glass-epoxy laminates and cores were hexagonal-cell honeycomb of either aluminium or glass-phenolic. For each beam, the lowest natural frequency, associated node locations, and the logarithmic decrement in free vibration were measured and compared with those predicted by four different theories.

26,761 **Flow-Induced Vibration and Noise in Tube-Bank Heat Exchangers Due to von Kármán Streets.** Chen, Y. N. *A.S.M.E., Paper No. 67 Vibr 48, presented* 29–31 Mar. 1967 [17 pp., 41 ref., 16 graphs, 13 diag.]

The frequency of vortex shedding from tubes and tube banks has been investigated by many workers, but their results have varied considerably. The present Author has previously correlated the existing data into a curve group, and now proceeds to further analysis of the problem. Results of experiments performed in a small wind tunnel are given, together with typical graphs. On this basis, some design proposals for suppressing vibration are made; in particular, that detuning baffles should be inserted into the tube bank.

26,762 **Statistical Energy Analysis of Vibrating Systems.** Ungar, E. E. *A.S.M.E., Paper No. 67-Vibr 8, presented* 29–31 Mar. 1967 [7 pp., 23 ref., 4 diag.]

Although the classical methods of vibration calculation are valid in principle at all frequencies, their use is very often impractical for high frequencies, particularly for randomly excited complex structures. A new approach is needed for dealing simply and effectively with high-frequency problems, such as those relating to sonically-induced fatigue, instrumentation performance, or noise transmission. The "statistical energy analysis" approach provides a relatively simple means for understanding and estimating the significant properties of multimodal random vibrations of complex systems; it permits treatment of complex vibration problems in terms of much simpler energy balances.

The first section of the paper reviews the concepts and properties of modes of structural vibrations, and points out some useful relations between modal response and total average response properties. The second section derives the basic relations that govern the exchange of energy between two coupled modes. The third section generalises this relation to permit determination of the flow of energy from one set of modes, representing one structure or fluid system, to another set of modes, representing another such system. The final section illustrates some applications of the statistical energy approach (response of indirectly excited systems; evaluating the effect of additional damping; interaction of sound and structures). Its use as a means of quantitative response prediction is at present hampered by the limited information available on coupling coefficients and damping in materials and structures.

26,763 **A Scale for the Degrees of Vibration Perceptibility and Annoyance.** SOLIMAN, J. I. *Ergonomics*, **11** (1968), p. 101 (Mar.) [22 pp., 61 ref., 7 tab., 11 graphs]

On the basis of a review of the numerous investigations published since 1900, criteria for vibrations permissible with regard to their effect on human beings (thresholds of perception and annoyance) are suggested. Dimensionless units for degrees of perceptibility and annoyance are introduced, and nomograms are given for determining these degrees from the frequency and displacement amplitude of the vibration. It is suggested that the criteria could serve as a basis for a British Standard.

CORROSION, FOULING, AND PREVENTION

26,764 **Conservation of Ship Bottoms with New Telsys Anti-fouling System.** SIEDENSTOPF, A., and ZACH, M. *Translation by the Chemical Translation Service, from Plaste Kautschuk*, **12** (1965), No. 8, p. 496 [10 pp., 8 ref.]

This paper consists mainly of an exposition of the general principles underlying the formulation and application of good anti-corrosive and anti-fouling paints for ships' bottoms. Mention is made of a new system [*Ed. note:* presumably East German] of which no details are given except that it comprises three anti-corrosive coats, one thick "isolative layer", and one anti-fouling layer, applied over a wash primer and of a total thickness of 0·15 mm (5·9 mil). Data obtained from a number of vessels treated with this system are given, and show that no appreciable loss of speed due to organic growths occurred for periods up to 34 months. Comparative tests showed that, with a conventional paint system, one vessel suffered a speed reduction of 4 knots in 6 months, whereas with the Telsys system no reduction occurred in one year. In another case comparable figures were a speed loss of 2 knots after 6½ months and no loss for 2 years.

Great importance is attached to the correct choice of the composition of the various layers of a paint system: the effectiveness of the system depends considerably on the interaction between the layers. The moisture to which marine paints are necessarily exposed is an important factor in the initiation of the processes that cause deterioration of the paint. Its effects can be reduced by lowering the permeability of the paint to corrosion products, and this depends, amongst other things, on the structure of binders and the character of the pigments.

The preparation and initial treatment of the steel are also very important. Sand or shot blasting is regarded as the best method of preparation, but care must be taken to ensure that the depth of the roughening is within certain limits. Peaks in the surface can act as sources of localised corrosion after a short period.

The prepared, rust-free steel surface should then be treated with a priming coat which, *inter alia*, eliminates residual salts such as chlorides or sulphates. The surface should also be thoroughly dry before the paint coating is applied. The pigments should be carefully chosen. For anti-corrosive properties they should include substances such as basic chromates or zinc oxide, while adhesion can be improved with vinyl resin wash primers.

Recently, some experience has been obtained of the effects of cathodic protection provided by special primers containing zinc or aluminium dust. It has been found that cathodic protection of this kind is not always compatible with the anti-fouling system. Good results have been obtained with zinc-dust pigments and anti-corrosive primers used with high-quality anti-fouling pigments, and also with zinc-spraying followed by painting. Zinc-dust pigments are approved only for freshly-blasted plates.

As regards application, the Authors state that the first coat of anti-corrosive paint must be applied by brush. Further coats can be sprayed, except the anti-fouling because of its toxicity. Rolling can also be used, but in most cases impairs the quality of the coating.

26,765 **A Fresh Look at the Mechanism of Corrosion in Boilers.** RIVERS, H. M. *A.S.M.E., Paper No. 65 WA BFS 1, presented* 7 11 Nov. 1965 [6 pp., 5 ref., 8 diag.]

Published block-specimen, hydrogen-effusion, and model-boiler experiments using mild steel have shown that, after a brief period of " flash " oxidation immediately following immersion in aqueous solution, corrosion proceeds either relatively slowly with the formation of a thin corrosion-resistant magnetite film, or rapidly with the development of a non-protective accumulation of iron oxide. In the latter case, film-destructive mechanisms induced experimentally by combinations of high temperature, stress, and quite high concentrations of hydroxide alkalinity or ferrous chloride have produced examples of severe metal loss, pitting, heavy oxide accumulation, and hydrogen damage; these are very similar to corrosion manifestations responsible for metal failure in real boilers.

OPERATION AND MAINTENANCE

26,766 **Moore-McCormack Plans Use of Computer " Dry Runs " for Service Patterns.** *Canadian Shipping and Marine Engineering News*, 39 (1967), p. 41 (Oct.) [½ p.]

This note draws attention to a statement by W. T. Moore, the chairman of Moore-McCormack Lines, Inc., concerning the success of a computer simulation model for planning the operations of one of his company's fleets (American Republics Line). On the basis of expected or proposed future conditions, the computer makes (in about one hour) a detailed analysis of a liner schedule involving 80 to 90 voyages in the course of a year. It prints out financial statements for each simulated period of operations, together with statistics on various matters of interest, such as cargo queues and ship utilisation. The model has proved so useful to the management that Moore-McCormack have contracted with IBM for the development of a more comprehensive one, covering all their cargo and passenger operations.

26,767 **Carrier Management the Vital Factor in Co-ordinated Land-Sea Transportation.** REEBIE, R. S. *A.S.M.E., Paper No. 67 TRAN 32, presented* 28 30 Aug. 1967 [7 pp., 2 diag.]

The systems approach to commercial logistics, which was first emphasised in the U.S. domestic economy, is now being widely applied to international operations. Thus ocean carriers may well find it necessary to develop management techniques similar to those which have proved effective for domestic road, rail, and air carriers. This applies particularly to routes where new shipping concepts may provide an excess of capacity, resulting in keener competition. Those carriers who adopt the most advanced management techniques are likely to be the most competitive. The Author discusses the subject in relation to business logistics, market management, market research, equipment and service planning, pricing services, organisational relationships, steps involved in industry marketing, personnel, forward planning, and operations management (with special reference to terminals).

26,768 **Cost Effectiveness in Pumping System Design and Operation.** BENZ, G. *J. Engng Power*, **89** (1967), p. 600 (Oct.) [5 pp., 10 ref., 4 tab.]

This is A.S.M.E. Paper No. 66 WA/FE-27, presented 27 Nov.–1 Dec. 1966.

A technique is presented for numerically evaluating pumping-system effectiveness by statistical analysis. The technique is applied to a hypothetical substitution of stainless-steel pump wear rings for bronze ones in the pumping station of an undertaking supplying river water to agricultural consumers. The objective is to find if the probable increase in quantity of water delivered annually compares favourably with the operational cost difference (taking account of down-time and maintenance) between the materials.

The cost-effectiveness model concerned is applicable to the great majority of maintainable systems, because these systems are known to experience constant failure rates and log-normal maintainability distributions. It can be used in planning optimum maintenance schedules. The accuracy of its predictions has been confirmed by comparison with published statistics for 113 hydraulic-turbine plants.

26,769 **M.A.N.'s Wear Data Recording System.** PAUER, W. *Shipbuild. Shipp. Rec.*, **112** (1968), p. 43 (12 July) [3 pp., 3 diag., 3 phot.]

M.A.N. have developed a system which, using an IBM 1232 Optical Mark Page Reader, facilitates the direct and automatic processing of records of cylinder-liner wear, taking into account all relevant factors. The type of wear records customarily kept by owners, yards, and engine builders may include all the necessary information, but to analyse such records by conventional manual methods to correlate wear with any one cause (e.g. type of lubricant) is a very lengthy and costly process.

M.A.N.'s new system replaces the usual record sheet by a special mark page which can be read by the IBM Page Reader. It contains space for 1,000 markings, each of which corresponds to a definite physical effect or property that may affect wear. The principal factors covered are the working conditions to which the liner is subjected, such as loading of the engine, piston speed, quality of combustion, etc.; relevant design and material factors, such as composition, heat treatment, and hardness of liner and piston-ring material; composition and quality of fuel and cylinder lubricant; duration of running time; and piston running-fit in liner.

The compilation of the mark page can be carried out by the engineers on board, using a special marking folder which the Author describes

He mentions also its application to two problems of interest to M.A.N., namely, the relation of wear rate to phosphorus content of the cylinder liner, and the correlation of wear rates in different engines.

26,770 **Preventive Maintenance of Equipment Subject to Continuous Deterioration and Stochastic Failure.** ROLL, Y., and NAOR, P. *Op. Res. Quart.*, **19** (1968), p. 61 (Mar.) [11 pp., 5 ref., 1 tab., 5 graphs]

This study is concerned with the derivation of a numerical procedure for determining the optimum policy regarding continuous attendance to, and preventive replacement of, equipment subject to both gradual deterioration and random breakdown. Some simple models are considered and, for special cases, explicit formulae are derived.

26,771 **Analysis of Machinery Noise as a Technique of Preventive Maintenance.** BOWEN, K. A., and GRAHAM, T. S. *A.S.M.E., Paper No. 67-Vibr-33, presented* 29-31 Mar. 1967 [3 pp., 1 ref., 2 tab., 2 graphs]

The Authors point out the advantages of noise and vibration measurements as a means of assessing deterioration of rotating machinery. They then describe a procedure developed at General Dynamics Corporation (Electric Boat Division) which allows relatively inexperienced machine operators to frequency-analyse machine noise for preventive-maintenance purposes. The equipment used is listed. The operator can compare his noise measurements with limit lines on spectral charts provided, to determine what repairs are required.

The procedure was devised primarily for the auxiliary machinery of submarines, but is generally applicable.

FIRE DETECTION AND PREVENTION

(*See Abstract No.* 26,718)

MISCELLANEOUS

26,772 **Desalination Problems and Techniques.** BOLTZ, C. L. *Publication issued by British Aqua Chem Ltd, London (1967)* [30 pp., 6 ref., 2 tab., 1 graph, 8 diag., 6 phot.]

This elementary review of the subject emphasises the merits of modern distillation techniques, in particular multi-stage flash distillation, for large plants.

26,773 **A Survey of Desalination by Reverse Osmosis.** KEILIN, B. *A.S.M.E., Paper No. 67-CNT-7, presented* 30 Apr.-3 May 1967 [8 pp., 17 ref., 3 tab., 3 graphs]

Although not yet in commercial operation, reverse-osmosis seems certain to become one of the leading desalination processes. This paper reviews the principles and development problems of the system, and its favourable cost-saving features. Some cost estimates are given for 1,000,000 gal/day plants handling (*a*) brackish water, (*b*) sewage water, and (*c*) sea water. Satisfactory single-stage membranes are available for (*a*) and (*b*), but not yet for (*c*).

See also Abstract No. 25,361 (May 1967)

26,774 **Water Desalination by Freezing.** BRIAN, P. L. T. *A.S.M.E., Paper No. 67 UNT 10, presented* 30 Apr. 3 May 1967 [14 pp., 19 ref., 2 tab., 3 graphs, 4 diag.]

Two " freeze-desalination " processes are currently under development: one uses direct contact between the brine and an immiscible liquid refrigerant (e.g. butane), whereas the other employs " vacuum flash " freezing. Both processes are described, and their relative merits compared from the technical and economic aspects. The various pilot plants which have been operated are discussed. The basic operations involved are ice crystallisation, separation of ice from brine, melting of ice by condensation of refrigerant, vapour compression, and feed product heat-exchange. These operations, and the equipment developed for performing them in the pilot plants, are considered in detail.

Although freezing processes are not nearly so well developed as distillation techniques, the estimated water costs are in general lower than for distillation, especially as regards plants of small or medium size (not more than a few million gallons per day).

Some account is also given of " hydrate " processes, in which a direct-contact refrigerant (e.g. propane, R12, or R31) also acts as a hydrating agent, so that the crystals formed are not ice but hydrate. These processes are in an early stage of development, but, in principle, offer thermodynamic and economic advantages over simple freezing.

26,775 **Operating Experience with a Large Flash-Type Sea-Water Distillation Plant.** STEINBRUCHEL, A. B., and BECK, H. *A.S.M.E., Paper No. 66-WA PTC 3, presented* 27 Nov.-1 Dec. 1966 [5 pp., 1 ref., 3 tab., 1 diag., 1 phot.]

This paper describes and presents data on an Aqua-Chem multi-stage flash distillation plant on the island of Aruba (Dutch West Indies); it is rated for 800,000 U.S. gallons of potable water per day. The plant is one of the first large flash-type plants to be operated continuously at elevated brine temperatures, using a scale-control system of continuous acid treatment followed by deaeration of the incoming sea water. The data relate to water and brine chemistry, effectiveness of scale and corrosion control, plant performance and operating costs, and maintenance requirements, during the first nine months of operation.

26,776 **Precipitated Impurities in Wet-Process Phosphoric Acid.** LEHR, J. R., FRAZIER, A. W., and SMITH, J. P. Reprint from *Journal of Agricultural and Food Chemistry (Washington, D.C.)*, **14** (1966), p. 27 (Jan. Feb.) [7 pp., 24 ref., 5 tab.]

26,777 **[Theory of the] Stability of Vertically-Rising Buoyancy-Propelled Bodies.** RODGERS, E. J. *A.S.M.E., Paper No. 66 WA UNT 14, presented* 27 Nov. 1 Dec. 1966 [8 pp., 4 ref., 11 graphs, 2 diag., 2 phot.]

See also item 25 in Abstract No. 25,656 (Sept. 1967).

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